THE ELEMENTARY PRINCIPLES
OF
MACHINE DESIGN,

EMBRACING THE PROPORTIONS OF
CONNECTING RODS, PISTON RODS AND PISTONS
FOR STEAM ENGINES,
COTTER JOINTS, SCREW WRENCHES, Etc., Etc.

With Full Instructions for Setting a Plain Slide Valve and Eccentric.

Also Practical and Explanatory Hints for Making all the necessary Calculations and Working Drawings.

By J. G. A. MEYER,

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HANDSOMELY ILLUSTRATED.

Simplicity and Accuracy are the Characteristics of this Work.

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PREFACE.

WHILE it is true that there are several very excellent works on machine design before the public, it is also unfortunately the case that we have no simple and cheap work which begins at the beginning and leads the student on by gradual and easy steps to an acquaintance with the more important departments of the subject.

The object of the writer in preparing the present little volume is to supply this want and to furnish a stepping stone, as it were, to the study of more elaborate works.

As the best way to impart a clear knowledge of any practical subject is to work out such easy examples as are likely to occur in every-day practice, the author has begun by giving the rules and directions required for the designing and drawing of one of the most common objects of the workshop—a screw wrench. Having worked out this easily understood example he then passes on to more intricate and difficult problems, and by degrees leads the young mechanic to the study of the slide valve and the eccentric.

Those who desire to carry their studies further in this direction will find ample material in the author's large work entitled, "Easy Lessons in Mechanical Drawing and Machine Design, arranged for Self-Instruction."

The very simplest language has been used throughout the volume, and the methods given are not only the most accurate but the simplest in use.

In the hope that this little book will lead many young mechanics to the successful study of the higher branches of the profession the author submits it to his fellow mechanics.

J. G. A. MEYER.

New York, 1897.

The Elementary Principles

OF

Machine Design.

By Machine Design is generally understood, those calculations and drawings which are required for the production of efficient, economical and attractive-looking tools and machines. This art demands such a combination of knowledge and skill as will insure a proper adjustment of strength to the work to be done, and such a distribution of the material employed and of the form into which it is put, that on the one hand there shall be no superfluous weight and consequent waste, and on the other no deficiency of material where material is needed. The machine designer begins where the inventor leaves off, and without the aid of a good designer most mechanical inventions would either prove failures or they would involve an expense for experimenting that would be ruinous.

It is true that very often we have parts of a machine or tools to design for which it is impossible to determine exactly the magnitude of the forces which act upon them. In such cases we must be satisfied with the results obtained by the application of empirical rules, that is, rules founded upon experiments or experience, and which do not depend on the strict application of the rules of higher mathematics, but on experience or observation alone.

From what just has been stated it will be seen that it is of the utmost importance that the young mechanic should make a thorough study of this branch of his art and the easiest and best way to gain a knowledge of the subject is to commence with some simple tool of every day use in the shop. Such a tool is the wrench, of which we will now give the proportions.

Proportions of a Wrench.

The wrench shown in *Figs. 1b* and *2b* is probably as good, useful and simple a subject as we can take up in these exercises. It will readily be understood that it is impossible to determine exactly the amount of pull which a workman or workmen may exert on the end of the wrench, or to compute the stresses set up in it by the rough usage to which it may be subjected. Hence we employ such empirical rules as will furnish us such proportions as have been found to give good results in practice. Furthermore, we will have the opportuuity of showing the advantage and the great simplicity of the use of symbolic expressions or formulas which are simply shorthand ways of writing a rule, and do not involve any knowledge of algebra, a science which is the great bugbear of mechanics. and is mistakenly considered to be a useless science in practice.

We may here point out that the advantage of these symbolic expressions is that they are applicable to designing any size of wrench. Thus, in finding the proportions of

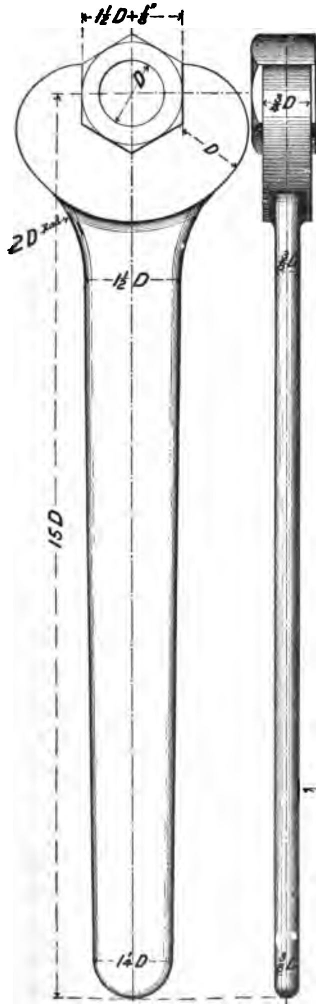
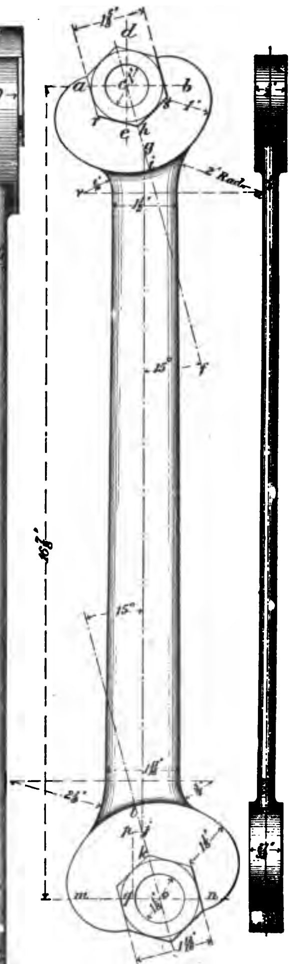


Fig. 1b.



J. G. A. MEYER.

Fig. 2b.

any part of the wrench we use the diameter of the bolt for which the wrench is to be designed, for a unit of measurement, and instead of writing "diameter of bolt" in full we simply write "D," which is the first letter of the word "diameter," and thereby not only save the labor of writing three words, but also add much to the simplicity with which the rule may be stated. If, now, these simple instructions are understood and kept in mind; and if, in addition to this, the student is familiar with the meaning of the ordinary signs used in arithmetic, we may say that he has mastered the whole subject of reading formulas.

Now, it seems to us that the student cannot fail to understand that *Fig. 1b* informs us that the distance between the jaws must be equal to $1\frac{1}{2}$ D + $\frac{1}{8}''$, which reads $1\frac{1}{2}$ times the diameter of the bolt plus $\frac{1}{8}$ inch; this dimension is the distance across the flats of the nut made in accordance with the American standard of bolt heads and nuts. The width of the jaw is equal to D; this indicates that the width must be equal to the diameter of the bolt. The width of the shank is marked $1\frac{1}{2}$ D, this indicates that the width must be equal to $1\frac{1}{2}$ times the diameter of the bolt. The radius of the arc joining the edges of the shank and the contour of the head is equal to 2D, or twice the diameter of the bolt; and the length of the wrench from the center of the bolt to the end is equal to 15 times the diameter of the bolt. The thickness of the wrench is plainly indicated in the side view in *Fig. 1b.* We may also point to the fact that in *Fig. 1b* the center line of the head of the wrench coincides with the center line of the shank.

2.—In *Fig. 2b* we show a wrench the head of which is set obliquely to the shank; here the center line of the head makes an angle of 15 degrees with that of the shank. With a wrench of this kind, a nut can be screwed up with the smallest possible angular movement of the wrench. Thus, taking hold of a hexagonal nut we can swing the wrench through an angular movement of 30 degrees, and turn the nut through $\frac{1}{12}$ of a revolution, and then reversing the wrench we can turn the nut through another $\frac{1}{12}$ of a revolution, and by repeating this operation we can turn the nut through any angle we please. If the nut had been a square one, the angle between the center line of the head and that of the shank should have been $22\frac{1}{2}$ degrees. Hence wrenches of this kind are suitable for screwing up nuts in contracted spaces.

3.—In order to explain the mode of procedure of making a drawing of the wrench shown in *Fig. 2b* we will take the following example.

EXAMPLE 1.—**It is required to make a drawing of a wrench for a bolt I inch diameter.**

Through any point *c* draw the center line *cf*, making an angle of 15 degrees with a vertical line; around the point *c*, and on the line *cf*, draw a hexagonal nut in accordance with the well-known methods given in all standard works on Mechanical Drawing. A simple and thorougly accurate method will be found in Chapter IV, of the Author's "Easy Lessons in Mechanical Drawing."

According to the proportions given in *Fig. 1b*, the distance across the flats of a hexagonal nut for a 1 inch bolt, and, consequently, the distance between the jaws of the wrench, will be $1\frac{5}{8}$ inches. Now, referring again to *Fig. 1b*, we notice that the outer contour of the jaws consists simply of arcs drawn from centers placed on the inner sides of the jaws; these lines must be produced so that the centers can be placed on these lines at a distance equal to the diameter of the bolt from the corner of the nut; this same distance is also equal to the radii of the arcs which form the outer contour of the jaws. Hence, from each of the centers *r* and *s* in *Fig. 2b*, and with a radius of 1 inch, draw arcs, and tangent to these draw, from the center *c*, an arc, and thus complete the whole contour of the head.

4.—Through the center *g*, of the width *hi*, draw a vertical line; this will be the center line of the shank, and its width will be, according to the proportions given in *Fig. 1b*, $1\frac{1}{2}$ inches; hence, at a distance of $\frac{3}{4}$ inch from the center line, and parallel to it, draw two straight lines, and join the line on the right hand side of the wrench and the contour of the head by an arc drawn with a radius of 2 inches; through the center *u*, from which this arc has been drawn, draw a horizontal line on which must be placed the center *v*, for drawing the arc joining the left hand edge of the shank and the contour of the head; this finishes the drawing of the upper end of the wrench.

5.—For the lower end we proceed as follows: Draw the horizontal center line *mn*; the distance between this line and the line *ab* should be equal to 15 diameters of the larger bolt, for which the wrench is to be designed; hence this distance will be equal to $15 \times 1\frac{1}{8} = 16\frac{7}{8}$ inches.

Now, for establishing the center *o* it will be best to make a separate drawing of the lower head, which will enable us to find the position of the point *j*, the center of the width *lk;* we then measure the center vertical distance from *j* to the center of the bolt, and make *pq*, in *Fig. 2b*, equal to it, and thus establish the position of point *j*; then through this point draw a center line making an angle of 15 degrees with the center of the shank, and cutting the line *mn* in the point *o*; around this point draw the nut and the contour of the head, following the same method as was used for drawing the upper head.

The width of the shank at the lower end will be greater than that at the upper end, which is designed for a smaller bolt; but the proportion $1\frac{1}{2}$ D, given in *Fig. 1b*, holds good in this case; since the diameter of the bolt at the lower end is $1\frac{1}{8}$ inches, the width of the shank will be $1\frac{1}{2} \times 1\frac{1}{8} = 1\frac{11}{16}$ inches. After the arcs at the lower end, joining the edges of the shank, and the contour of the head have been drawn, join these arcs and the upper ones by straight lines drawn tangent to the arcs. If there had been no head at the lower end the width of the shank would have been $1\frac{1}{4} \times 1 = 1\frac{1}{4}$ inches.

For drawing the side view so as to show the thickness of the wrench, commence by drawing the center line, and around this draw an outline of the wrench in accordance with the proportions given in *Fig. 1a*. A further explanation is unnecessary.

Proportions of Cottered Joints.

6.—The proportions of cottered joints are given in symbols on the following illustrations. These symbolic expressions will be given in ordinary language as we proceed. It should, however, be understood that the magnitude of the forces acting upon these joints can generally be computed, and, therefore, not all of these symbolic expressions should be looked upon as empirical rules; indeed, many of them have been deduced from sound reasoning based upon the principles relating to the strength of materials. But at this early stage the student may not be sufficiently advanced to understand this reasoning, consequently we will, for the present, explain only the manner of applying these formulas to the designing of joints of this kind, and work out the following example. This subject is further considered in the author's "Easy Lessons in Mechanical Drawing."

EXAMPLE 2.—**In** *Fig. 3b* **the rod is subjected to a pull or tension of 25,000 pounds; it is made of wrought iron; find the diameter of the rod and dimensions of the cottered joint.**

Our first step will be to find the diameter d of the rod. In ordinary practice we can allow a pull or tension of from 10,000 to 12,000 pounds per square inch of cross-section of the rod, depending on the quality of the iron. Let us adopt 10,000 pounds per square inch for the safe tension on the cross-section of the rod. Now, it will not require much mental labor to conceive that if each square inch of section can stand a tension of 10,000 pounds, then for a tension of 25,000 we will require an area of $\frac{25000}{10000} = 2\frac{1}{2}$ square inches; and as soon as this area of the cross-section is known we can easily find the diameter of the rod by the ordinary rules of mensuration. But the draftsman or designer would never think of spending so much time and labor in computing this diameter; he takes, so to speak, a short cut and a much easier way of finding this diameter by referring to a table of areas of circles found in all engineers' pocketbooks and thus finds at once that a circular cross-section containing the $2\frac{1}{2}$ square inches will be nearly $1\frac{13}{16}$ inches diameter, or in decimals, which are easier to use in the following computations, the diameter of the circle will be 1.8125 inches. The manner of handling decimals is explained in our regular course of instruction.

The end of the rod through which the cotter passes is weakened by the slot, and in order to make the end equal in strength to the body we must increase the diameter at the end, and this increase can be computed without expending much labor by the formula given on the illustration.

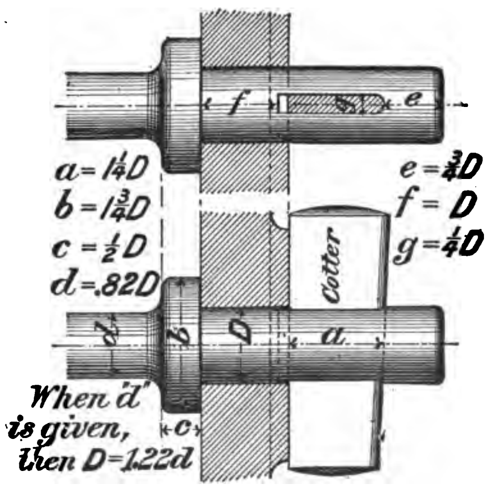


Fig. 3b.

Now, in all these formulas the letter D stands for the diameter of the end, and the other letters refer to the parts having the same letters. Thus we see that $D = 1.22d$, which reads, the diameter at the end is equal to 1.22 times the diameter *d* of the body of the rod; the diameter *d* we have found to be equal 1.8125 inches; hence the diameter D must be equal to

$$1.22 \times 1.8125 = 2.21125, \text{ say } 2\tfrac{7}{32} \text{ inches.}$$

It will also be noticed that this diameter D is the unit of measurement in all the formulas; thus we see that the width a of the cotter near the center is equal to $1\frac{1}{4}$ times the diameter D; hence $a = 1\frac{1}{4} \times 2.21125 = 2.764$, say $2\frac{3}{4}$ inches for the width of the cotter near the center; the diameter b of the collar is equal to $1\frac{3}{4}$ times D, hence we have $1\frac{3}{4} \times 2.21125 = 3.869$, say $3\frac{7}{8}$ inches for the diameter of the collar; the thickness c of the collar is equal $\frac{1}{2}$ D, or $\frac{2.21125}{2}$ = say $1\frac{1}{8}$ inches. The distance e from the cotter to the end of the rod is $\frac{3}{4}$ times D, hence this distance will be equal to $\frac{3}{4} \times 2.21125$ = say $1\frac{22}{32}$ inches. The distance f from the slot to the collar depends on the thickness of the part through which the end of the rod passes; and when this part is of cast iron, and if it is first to be found, it is often advisable to make it suitable to receive a rod in which the distance f is equal to the diameter of the end. The thickness of the cotter in joints of this kind is generally equal to $\frac{1}{4}$ the diameter D; hence, in this case, this thickness is equal to $\frac{1}{4} \times 2.21125 = \frac{9}{16}$ inch. The bearing surface of the cotter against the part to which the rod is attached should not be less than the bearing surface of the cotter against the end of the rod; according to this, the length of the cotter should be equal to twice the diameter D; but in practice this length is generally greater, so that the cotter can be driven home before the end of the cotter reaches the rod. The taper of the cotter should be about $\frac{1}{4}$ inch in 12 inches.

7.—In *Fig. 4b* we have given several views of a cottered joint, such as is very often used for connecting a locomotive valve-stem whose diameter is d and a valve-rod whose diameter is d_2.

In designs of this kind it is difficult to determine exactly the amount of pull or tension on the stem; but for the purpose of making these lessons useful and practical, so that the rules can be applied to similar joints used for different purposes, we will assume that the total tension on the valve-stem is 2,500 pounds.

In looking at the general construction of this joint, it is easily seen that the tapered shank through which the cotter passes is the weakest part; and, furthermore, the weakest portion of the shank is at the smallest diameter through which the cotter passes; this part is marked D, and of this we must first find the area of the cross-section required for resisting the given total tension of 2,500 pounds. In computing this we must not overlook the fact that when this joint is used for connecting a valve-rod and stem the joint will be subjected alternately to tension and compression, and the rapid change from one kind of stress to the other is

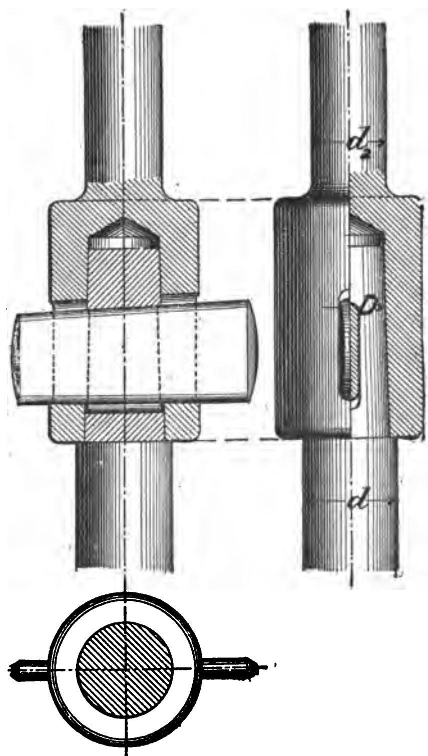


Fig. 4b.

liable to produce shocks, and for a material subjected to shocks we must use a larger factor of safety than was employed in the last example, in which the rod was assumed to be subjected to a steady load, or to tension only. Hence, instead of allowing a safe working stress of 10,000 pounds per square inch we must not allow more than 5,000 pounds to the square inch. Taking this allowable working stress it will readily be seen that for a total stress of 2,500 pounds we will need a cross-sectional area of $\frac{2500}{5000} = \frac{1}{2}$ square inch, without any allowance for the cotter hole, and the diameter of a circle containing an area of $\frac{1}{2}$ square inch is very nearly equal to $\frac{13}{16}$ inch.

Comparing this mode of procedure with that of computing d in the last example, we see that we have here computed the diameter d_2 of the valve rod under the assumption that it is subjected to tension and compression; hence, for computing the diameter D in *Fig. 4b* so that the proper allowance is made for the metal cut out for the cotter, whose thickness is to be $\frac{1}{4}$ of D, we follow the rule given in *Fig. 3b*, namely, multiply $\frac{13}{16}$, or the equivalent decimal, which is .8125, by 1.22, and we thus get

$$.8125 \times 1.22 = .991 \text{ say 1 inch,}$$

for the diameter D.

Now, to find the dimensions of some of the other parts of this joint we use the rules given in *Fig. 3b*. Thus, the width of the cotter will be $1\frac{1}{4}$ times D; and since D is equal to 1 inch we have for this width, $1\frac{1}{4} \times 1 = 1\frac{1}{4}$ inches.

For the thickness of the cotter we have $\frac{1}{4}$ D, which is equal to $\frac{1}{4}$ inch. For the distance from the cotter to the end of the shank we have $\frac{3}{4}$ D; hence this distance is equal to $\frac{3}{4}$ inch; many engineers make it equal to D. On account of enlargement of the hole in the end of the socket, we should make the distance from the cotter to the shoulder not less than $\frac{3}{4}$ D. The outside diameter of the socket is generally made equal to twice D; hence, in this case, the outside diameter of the socket will be 2 inches.

We have now all the dimensions required for making a drawing of the joint.

In making the drawing, commence by drawing the horizontal center line, and on it set off the length of the shank and width of the cotter; in fact, draw a section of the cotter in accordance with the dimensions as computed above; this will establish the position of the diameter D, which, as we have seen, is to be 1 inch. Through the extremities of this diameter draw the straight lines indicating the taper of the shank; this taper, or the variation in the diameter of the shank, is generally equal to $\frac{1}{8}$ inch in 2 inches, or 1 in 16, and in some cases it is 1 in 32. In this way we establish the large diameter of the shank, then allowing say $\frac{1}{16}$ inch for a shoulder around it, we obtain the diameter d of the stem, which is the last dimension to be determined, for the completion of the joint.

In this design a collar on the stem is not permissible, because the diameter of stem must be uniform through the whole of its length, so that it can pass through the stuffing box gland and work in it with a fairly good fit.

It would seem that the diameter d_2 should be equal to .82 times D, as given in the formula in *Fig. 3b.* But since we have assumed that this joint is one for connecting a locomotive valve rod and is subjected to a bending action, the diameter d_2 should not be less than D.

8.—But in cases where a cottered joint is used for other purposes, a collar on the stem may be used, as shown in *Fig. 5b*; and if the shank in the socket is tapered, the dimensions of the collar are found by the formulas given in *Fig. 5b*. If the shank is of uniform diameter throughout, we prefer to compute the dimensions of the collar by the formulas given in *Fig. 3b*. The distance $f = \frac{1}{2}$ D given in *Fig. 5b* is the smallest per-

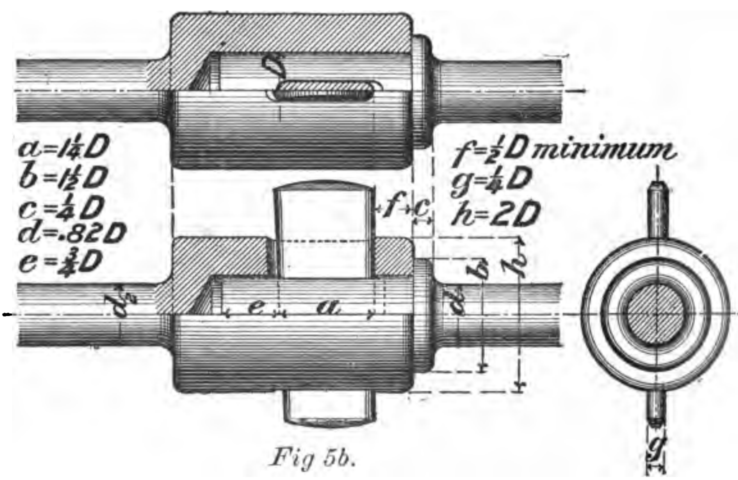


Fig 5b.

missible; and whenever the design will permit, we prefer to make it equal to $\frac{3}{4}$ D or D. In regard to computing the dimensions of the joint nothing need be said, as the formulas given in *Fig. 5b* are used in precisely the same way as explained above.

9.—When extreme lightness is demanded in a design, the weight of the socket may be lessened by reducing its diameter b_2, as shown in *Fig. 6b*, and in practice it is generally made equal to 1.5D; but in accordance with theoretical reasoning it be could made equal to 1.34 D. Between the cotter and the collar the outer diameter h of the socket cannot be reduced without reducing the bearing surface of the cotter against the socket, and this is objectionable; hence the diameter h is generally equal to 2D.

10.—In *Fig. 7b* we have shown several views of a knuckle-joint for a locomotive valve rod; with this kind of joint the valve rod will not be subjected to any bending action, and, therefore, the diameter d_2 at the end of the rod can be made equal to .82 times the diameter D of the weakest part of the shank. Of course the diameter D

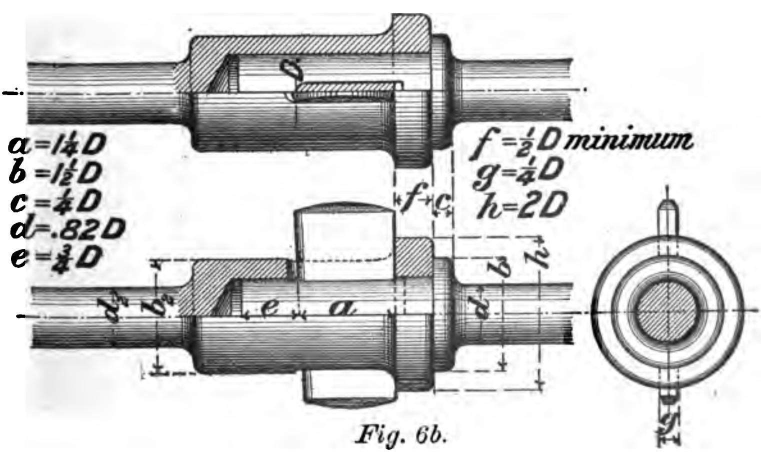


Fig. 6b.

must first be computed, as explained in the preceding example. The drawing shows a construction well adapted to protecting the pin from dust.

11.—All the preceding illustrations represent cottered joints made wholly of wrought iron. *Fig. 8b* shows

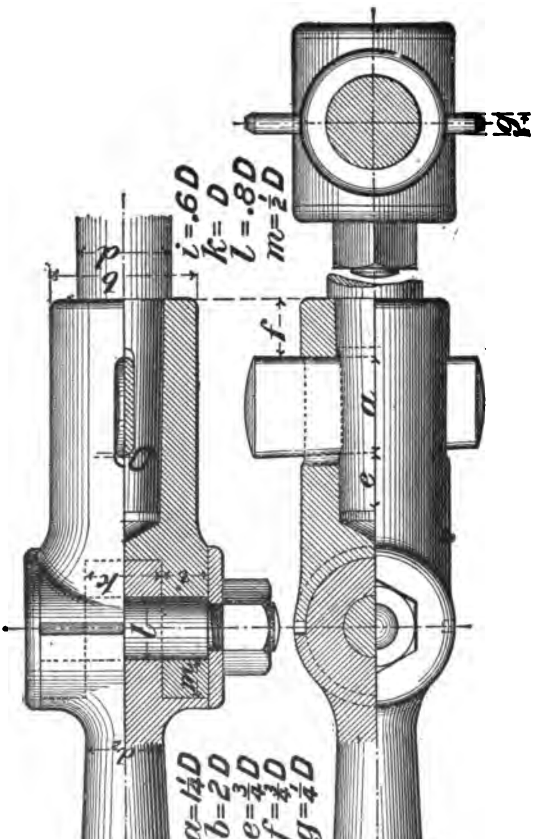


Fig. 7b.

a wrought iron rod cottered to a cast iron hub. In this case we should make the outside diameter of the hub equal to twice the largest diameter of the shank. For finding the other dimension of the rod and hub we use the same rules as given in the previous examples relating to cottered joints.

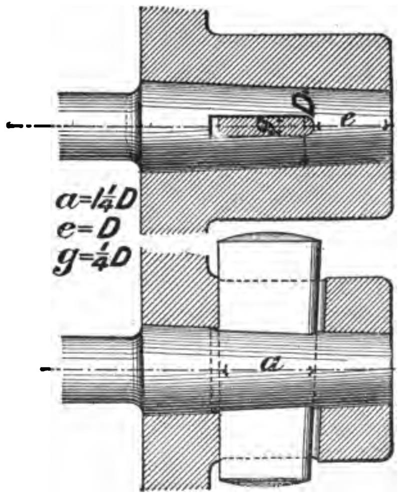


Fig. 8b.

12.—*Fig. 9b* gives the proportions of a connecting rod strap. The width B of the strap is generally made equal to, or a very little larger than, the diameter of the rod; and when the width B is established the thickness *b* of the strap can be computed. In designing a connecting

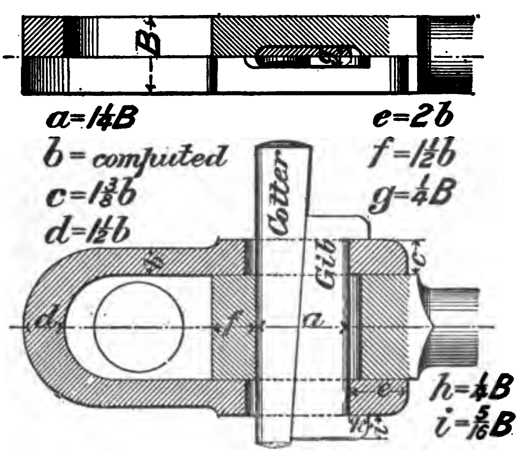


Fig. 9b.

rod the first dimension computed is its diameter; but for the sake of simplicity we will assume that this diameter and, consequently, the width B which, in this case, is equal to it, is known.

EXAMPLE 3.—**It is required to find the dimensions of a connecting rod strap of the form shown in Fig. 9b. The connecting rod is to be used for an engine with a cylinder 4 inches in diameter. The total tension or pull on the strap is 1,300 pounds. The diameter at the neck of the connecting rod is 3/4 inch and width B of the strap is 3/4 inch also.**

Here, again, we may adopt a safe working stress of 5,000 pounds per square inch, and therefore the aggregate area of both cross-sections of the thinnest part *b* of the strap will have to be

$$\frac{1300}{5000} = 0.26 \text{ square inch,}$$

and the area of one section of the strap, that is a section through *b*, will have to be

$$\frac{0.26}{2} = 0.13 \text{ square inch.}$$

Now, dividing this by the width B of the strap, which is $\frac{3}{4}$ inch, we have

$$\frac{0.13}{0.75} = 0.17, \text{ say } \tfrac{11}{64} \text{ inch}$$

for the thickness *b* of the strap. This thickness is required for resisting the pull or tension on the strap, and does not meet other demands of a practical nature, for which an allowance must be made. The brasses are liable to wear unevenly in the inner surfaces of the strap, thereby reducing its strength, which is not permissible; we also need a little extra thickness for trueing up, and therefore we must add $\frac{1}{8}$ inch to the thickness found by computation as above, and we thus get for the thickness b

$$\tfrac{11}{64}'' + \tfrac{1}{8}'' = \tfrac{19}{64} \text{ say } \tfrac{5}{16} \text{ inch.}$$

As soon as the thickness b has been determined we will have no difficulty, with the aid of the formulas given in the illustration, to compute the other dimensions. For instance, the thickness c of that part of the strap which has been weakened by the cotter hole must be reinforced by making this thickness equal to $1\tfrac{3}{8}$ times b, as indicated by the formula, and we thus get

$$c = 1\tfrac{3}{8} \times \tfrac{5}{16} \text{ say } \tfrac{7}{16} \text{ inch.}$$

The thickness d is equal to $1\tfrac{1}{2}$ times b, hence we have

$$d = 1\tfrac{1}{2} \times \tfrac{5}{16} = \tfrac{15}{32} \text{ inch.}$$

The distance e from the end of the strap to the gib is twice b hence for this distance we have

$$e = 2 \times \tfrac{5}{16} = \tfrac{5}{8} \text{ inch.}$$

In a similar way we find all the other dimensions with the aid of the given formulas.

13.—In this example we have a cotter and gib combined. The purpose of the gib is to prevent the ends of the strap from spreading when the cotter is driven in.

With the gib we have also the advantage of giving a greater bearing surface for the cotter to slide on. In order to give the same amount of bearing surface on each side of the cotter two gibs are used, see *Fig. 10b*.

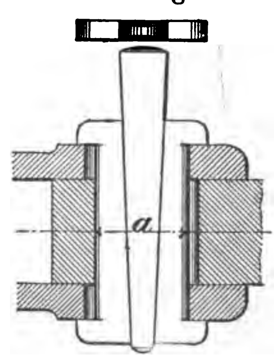


Fig. 10b.

14.—The total width a of the

cotter and gib shown in *Fig. 9b*, or the total width *a* of the cotter and two gibs, as shown in *Fig. 10b*, is equal to that of a cotter used by itself; that is to say, it is equal $1\frac{1}{4}$ times the width B of the strap; but it should be remembered that this width is suitable only when the thickness of the cotter is equal to $\frac{1}{5}$th of its width, or when the thickness g is equal to $\frac{1}{4}$th of the width of the strap. These proportions agree very well with practice. The taper of cotters of this kind is generally $\frac{5}{8}$ inch to $1\frac{1}{4}$ inch in 12 inches.

For great tapers some means must be provided for preventing the cotter from slipping out of place. Sometimes a set-screw tapped into the side of the end of the connecting rod is employed; sometimes a screw is formed on the end of the gib, as shown in *Fig. 11b*. When this screw is parallel to the perpendicular edge of the gib a slot must be cut through the end of the cotter, as shown in *Fig. 11b*, so as to allow some play when the distance between the parallel sides of the cotter and gib increases.

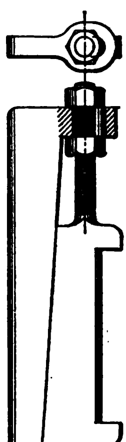


Fig. 11b.

When the screw is parallel to the tapered side of the gib, no slot at the end of the cotter will be required; a round hole giving the screw sufficient play to pass through it is all that is needed.

15.—When a cotter alone is used, one end of the slot will have to be tapered to suit the taper of the cotter; but when a cotter and gib are used, the ends of the slot are parallel. It has been found to be good practice to make the ends of slots of a semicircular form, and the edges of the cotter and of gib to suit these forms. It will save not only labor, but will also add strength to the connecting rod strap, because when the ends of the slots are of a square form we have sharp corners, and from these a crack is liable to start and extend crossways of the strap and completely ruin it.

16.—No regular rule can be given for determining the smallest width of the cotters, such as are shown in *Figs. 9b*, *10b* and *11b*; as a guide in designing it might be said that this width should not be less than the thickness of the cotters; frequently it is a little greater. The length of these cotters also depends on individual preference, but generally they should be sufficiently long to take up as much of the play between the brasses as is possible without being compelled to use liners between the brass and the butt end of the rod; hence it will be seen that the length of the cotter also depends on its taper.

17.—Cotters are generally driven in by hammering; but sometimes we meet with cases in which the limited space for the cotter prevents not only the use of the hammer, but also compels the use of a comparatively short cotter with a great taper. For such cases the form shown in *Fig. 12b* is well adapted. The screw at the end is suitable for moving the cotter and holding it in place; otherwise, the taper when greater than 1 in 7 would cause it to fly out.

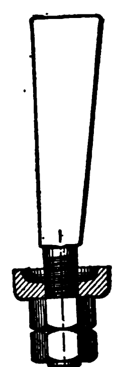


Fig. 12b.

18.—Usually there is no difference in the dimensions of a wrought iron or steel cotter; but generally the latter material is preferred, because a steel cotter will withstand hammering better than a wrought iron one.

19.—*Fig. 13b* shows three views of a knuckle joint connecting the rods *R* and *S*. The rod *R* has a forked end, and the rod *S* has an eye forged to it. This design is not adapted to all kinds of valve rods, because when the rod *S* is to be used for a valve stem the eye at its end will prevent it from passing through the gland of the stuffing box; similar remarks apply to the rod *R* if this is to be used for a valve stem. In order to make this joint suitable for connecting a valve stem and rod it will be imperative, in many cases, to forge separately the eye on the rod *S*, or the forked end separately on the rod *R*, whichever is to to be used for a valve stem, and cotter it to the rod, as shown in *Fig. 7b*. However, for many purposes this joint is an excellent one. The diameters of the rods *R* and *S* are, of course, equal to each other, and these diameters *D* must first be computed. The pin *p* through the eye is subjected to a shearing stress, and if this stress alone is considered we can make the diameter *d* of this pin equal to $\frac{8}{10}$ of the diameter *D* of the rod or, as indicated by the formula accompaning the design: $d = .8D$, in which $d =$ diameter of the pin and $D =$ diameters of the rods. But if the pin fits loosely in the eye then it will not only be subjected to a shearing stress, but it will also be subjected to a bending action, and to resist these combined stresses the diameter *d* of the pin should be equal to $.9D$. Furthermore, in order to obtain a larger bearing surface, the diameter of this pin is often increased to $1.2\,D$, so that in practice we find that the diameter *d* varies from $.8D$ to $1.2D$. The width *A* of the eye is generally equal to, or slightly greater than, the diameter *D*; but if the motion of the eye around the pin is considerable, then in order to obtain a larger bearing surface, the width *A* is increased to $1\frac{1}{4}D$, and sometimes even greater than this. The pin *p* is prevented from turning by a small pin driven into one of the jaws as shown on the drawing; again, to prevent a too rapid wear, the pin *p* is often casehardened, and a casehardened bushing is driven into the eye. The thickness of the bushing depends on the diameter of the pin *p*; for a pin 1 inch diameter the thickness of the bushing may be $\frac{1}{8}$ inch; for larger pins this thickness is slightly increased. It should also be noticed that the arc joining the inner surfaces of the jaws of the fork is not drawn from the same center from which the outer arcs are drawn, thereby increasing the thickness of

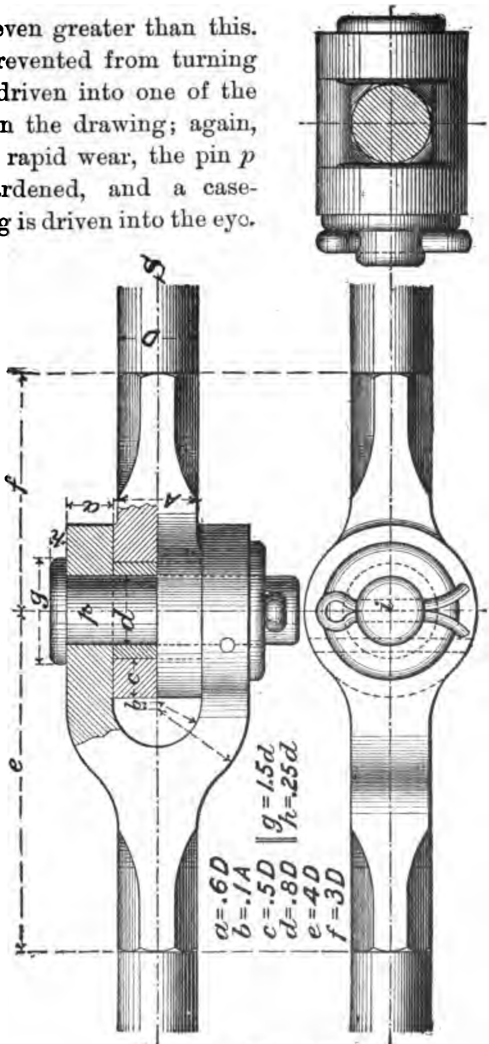


Fig. 13b.

the jaws towards the center of the rod, and thus giving the jaws a greater strength to resist those forces which act with a greater leverage. Hence the distance b between these centers is equal to $\frac{1}{10}$ of the width A of the eye, or as indicated by the formula, $b = .1A$.

Before making a drawing of this joint it will be best to compute all its dimensions, and since these computations are similar to those given in Articles 6 and 7, we shall be very brief in working out the following example.

EXAMPLE 4.—**It is required to find the dimensions of a knuckle joint of a form as is shown in *Fig. 13b*; it is subjected alternately to tension and compression; the total load is 3,000 pounds.**

Since the joint is subjected to tension and compression we allow a safe working stress of 5,000 pounds per square inch; hence the cross-sectional area of the rod R or the rod S will be

$$\frac{3,000}{5,000} = .6 \text{ square inch},$$

and the diameter of a circle containing .6 square inch is $\frac{7}{8}$ inch, and this, of course, will be the diameter of the rod R or the rod S. The following computations will become easier by using the equivalent decimal for the fraction $\frac{7}{8}$: it is .875. Now, to make as plain as possible the manner of using the symbolic expressions given in connection with *Fig. 13b*, we will write under each letter its numerical value, thereby showing the great simplicity of the whole subject. An inspection of the formula shows at once that the diameter D of the rod is the unit of measurement. Now, for the thickness a of the jaw we have

$$.6 \times D = a$$
$$.6 \times .875 = .5250 \text{ inch.}$$

The thickness A of the eye we can make 1 inch; hence, the distance b between the centers of the arcs of the jaw will be determined by the formula

$$.1 \times A = b$$

which gives us

$$.1 \times 1 = \cdot 1 \text{ inch.}$$

For the thickness c of the eye we have

$$.5 \times D = c$$
$$.5 \times .875 = .4375 \text{ inch.}$$

For the diameter d of the pin p we have

$$.8 \times D = d$$
$$.8 \times .875 = .7000, \text{ say } \tfrac{23}{32} \text{ inch.}$$

For the distance e, from the center of the jaw to the end of the octagon at finish of the forked end, we have

$$4 \times D = e$$
$$4 \times .875 = 3.5$$

For the distance f, from the center of the jaw to the end of the octagonal finish of the eye, we have

$$3 \times D = f$$
$$3 \times .875 = 2.625 = 2\tfrac{5}{8} \text{ inches.}$$

For the diameter g of the head of the pin we have

$$1.5 \times d = g$$
$$1.5 \times .7 = 1.05, \text{ say } 1 \text{ inch,}$$

and for the thickness h of the head of the pin we have

$$.25 \times d = h$$
$$.25 \times 7 = .175, \text{ say } \tfrac{3}{16} \text{ inch.}$$

Now, having determined by computations all these dimensions, there should be no difficulty in making the drawing of the joint. Commence by drawing first all the center lines, and around these lay off symmetrically all the dimensions as found above, and take care not to violate the rules of projection; for instance, the center line of the pin in the upper view or plan and the center *i* of the same pin in the lower view or elevation must lie in one straight vertical line; this will bring the end of the octagonal finish of the eye in the plan and elevation in one straight line; the same remarks apply to the finish of the forked end. The horizontal center lines in the end view and elevation must also lie in the same horizontal line; in fact, the rules of projection given in our regular lessons must be adhered to.

Double Nut Joint.

20.—*Fig. 14b* shows a double nut joint; it is designed for the purpose of adjusting accurately the length of a rod which is made in two parts; one part has a right-handed thread on its end and the other part has a left-handed thread, so that when the nut is turned in one direction it will increase the total length of the rod, and when turned in the opposite direction the total length of the rod will be shortened.

The weakest part of the rod is at the bottom of the thread, consequently, in computing the strength of this rod we must find the area of this section; working out the following example will make the whole matter plain.

EXAMPLE 5b.—Find the dimensions of a double nut joint of the form shown in *Fig. 14b*. The rod is made of wrought iron and is subjected to a tension and compression of 6,000 pounds.

Since the rod is subjected to tension and compression, we again adopt a safe working stress of 5,000 pounds to the square inch; consequently, the area of the section at the bottom of the thread will have to be

$$\frac{6{,}000}{5{,}000} = 1.2 \text{ square inches.}$$

Now, referring to our illustrated table of standard threads, bolt heads and nuts, we find that the diameter outside of the thread corresponding to an area of 1.2 square inches at the bottom of the thread is $1\frac{1}{2}$ inches; therefore the diameter D of the rod will be $1\frac{1}{2}$ inches; and this diameter will be the unit of measurement for determining the dimensions of the nut, thus:

The distance across the flats of the nut is to be $1\frac{1}{2}$ times the diameter D of the rod, plus $\frac{1}{8}$ inch, which gives us

$1.5 \times D + \frac{1}{8}$ inch

$1.5 \times 1.5 + \frac{1}{8}$ inch $= 2\frac{3}{8}$ inches.

The length of the hexagonal part c of the nut is

$1.5 \times D = c,$

$1.5 \times 1.5 = 2\frac{1}{4}$ inches.

In a similar simple way we find all the other dimensions of the nut

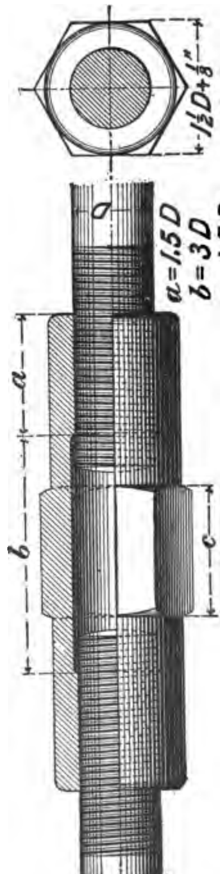


Fig. 14b.

by means of the formulas given on the drawing. These examples show the simplicity of the use of formulas.

The half of the nut above the center line is shown in section, but it is not absolutely necessary to do so on a working drawing; we have done so here simply for the purpose of familiarizing the student with showing things in section when it is necessary to do so.

Connecting Rods.

21.—A link which connects a rotating crank with a reciprocating piece is called a connecting rod. For instance, the crank of a steam engine and the crosshead are connected with a connecting rod. A link which connects two rotating cranks is called a coupling rod. Thus, the rod which connects the crank pins of two driving wheels on a locomotive is a coupling rod. In American practice we call it a side rod.

22.—Connecting rods are made of various forms. For engines running at a moderate speed a connecting rod of circular cross-section is generally used. In order to make the rod as light as possible without much reducing its resistance to bending, it is tapered from the center towards the ends, as shown in *Fig. 15b,* or it is tapered from the center to the crosshead end only, and from the center to the crank pin end it is parallel or cylindrical in form, as shown in *Fig. 16b.* In some cases the rods have a uniform taper from end to end; but the thickness of the rod remains uniform by cutting flats on it sides, as shown in *Fig. 17b.* For high speed engines, the connecting rods have a section of rectangular, or approximately rectangular form; their width is uniformly tapered from end to end, the thickness remaining the same throughout. See *Fig.18b.*

The length of a connecting rod is always measured from the center of the crank pin to the center of the crosshead pin.

23.—When the engine starts slowly we may neglect the inertia forces of the connecting rod, but its actual pull or thrust must be computed and provided for. There are other forces which the connecting rod must resist; but the magnitude of these forces is often difficult, if not impossible, to compute. For instance, sometimes water will collect in the cylinder, and it will be impossible to estimate the forces due to the presence of water in the cylinder. Again, when the cut-off occurs later than half stroke, the pull or thrust will at times during one revolution of the crank be greater than that on the piston rod, but this increase can be found graphically by the following simple diagram, which will also give us the pressure of the cross-

D = diam. of cyl. in inches.
p = pressure per square inch on piston.

$d = \frac{D}{55}\sqrt{p}$

$d_2 = \frac{D}{50}\sqrt{p}$

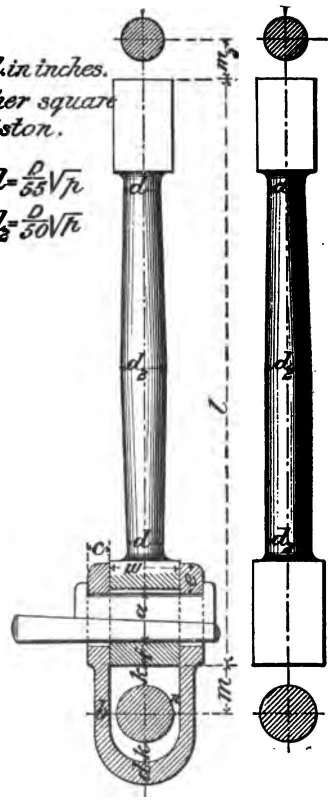


Fig. 15b. *Fig. 16b.*

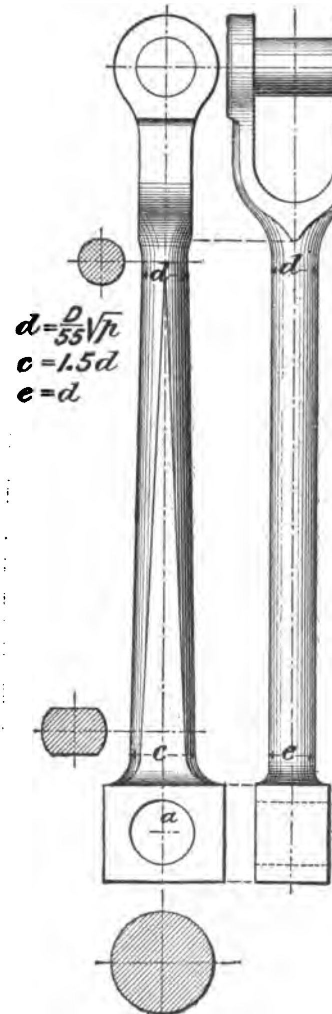


Fig. 17b.

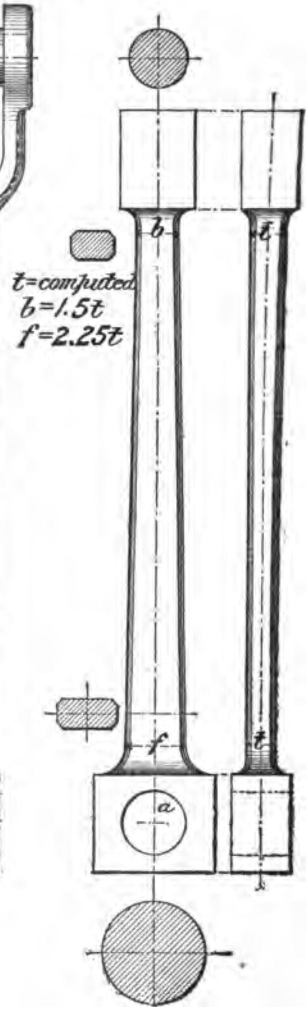


Fig. 18b.

head against the slides, which must be known before we can compute the dimensions of the slides or guides.

24.—*Fig. 18½b.*—Draw the horizontal line AC, and through any point C on this line draw a perpendicular CB and make its length equal to the length of the crank; from B as a center and with a radius equal to the length of the connecting rod draw an arc cutting the horizontal line AC in the point A; join A and B by a straight line and thus complete the triangle ABC. The side AB will represent the pull or thrust on the connecting rod; the side AC will represent the pull or thrust on the piston rod, and the side BC will represent the pressure of the crosshead against the slide. Now, the pull or push of the connecting rod will

Fig. 18½b.

be as much greater than that of the piston rod as the line AB is longer than the line AC. From this statement it will be seen that in order to find the numerical value of these stresses we need to apply only the simple rules of proportion. To make this whole matter plain we will work out the following example:

EXAMPLE 5.—**Find the thrust of a connecting rod for an engine having a cylinder 4 inches diameter; stroke, 6 inches; initial steam pressure, 100 pounds per square inch; the length of the connecting rod is to be 2½ times the stroke.**

According to the above conditions the length of the connecting rod will be

$$2\tfrac{1}{2} \times 6 = 15 \text{ inches.}$$

Draw the lines AC and CB perpendicular to each other, and make CB equal to the length of the crank, namely, 3 inches; from the point B as a center and with a radius equal to the length of the connecting rod, 15 inches, draw an arc cutting the line AC in the point A; join A and B, and thus complete the right-angled triangle ABC.

By measurement it will be found that the line AC is 14.69 inches long; this length represents the total load on the piston rod. For computing this total load in pounds we must multiply the area in square inches of the piston (12.56 square inches) by the steam pressure per square inch, namely, 100 pounds, and thus we get

$12.56 \times 100 = 1{,}256$ pounds for the total load or thrust on the piston rod,

and this load is represented by the line AC, or by the length of 14.69 inches.

Since the thrust on the connecting rod is as much greater than that on the piston rod as the line AB is greater than the line AC, and since the line AC represents in this case 1,256 pounds we can easily find the thrust in pounds on the connecting rod represented by the line AB by the following simple statement in proportion.

$$14.69 : 15 :: 1256 : x;$$

this gives us

$$\frac{15 \times 1256}{14.69} = 1282.5 \text{ pounds.}$$

From this it will be seen that the difference between the pull on the connecting rod and that on the piston rod is very small; this is generally the case in all engines, and, therefore, this difference is usually neglected. But the empirical rules or formulas used for computing the dimensions of the connecting rod will allow for this small difference, and for the forces acting on the rod, which cannot be estimated or accurately determined. It may therefore seem that the diagram in *Fig. 18½b* is useless in designing an engine, and so it is as far as the thrust on the connecting rod is concerned; but when we have to find the pressure of the crosshead against the slide it is very useful; thus, we have seen that the side BC of the triangle *Fig. 18½b* represents this pressure, which is as much less than the pull on the piston rod as the side BC is shorter than the side AC; we therefore employ the simple rule of proportion for finding the numerical value of this pressure; thus remembering that the total load on the piston is 1256 pounds, we have

$$14.69 : 3 :: 1256 : x;$$

and this gives us

$$\frac{3 \times 1256}{14.69} = 256.5 \text{ pounds}$$

for pressure of the crosshead against the slide.

25.—This diagram also serves to show that the longer we make the connecting rod the less will be the pressure of the crosshead against the slide. But the length of the connecting rod is also affected by the limited space which the engine is to occupy, consequently, the length of connecting rods has the following proportion:

For stationary engines, the length of the connecting rod is generally equal to from 2.5 to 3 times the length of stroke; in some engines this length is exceeded and made as long as 3.75 times the length of the stroke; and occasionally we find it to be as short as twice the length of the stroke. In marine engines, which very often must occupy as small space as possible, the length of the connecting rod is as short as 1.75 times the stroke. In locomotive engines the length depends on the relative position of the driving wheels, and is often much longer than given above.

26.—Another thing which the diagram *Fig. 18½b* serves to show is the direction in which the crank should turn. For engines which always run in one direction the pressure on the slides should be due to a push on the connecting rod when the crank pin is above the slide, as shown in *Fig 18½b*; under these conditions the crosshead will be pushed towards the slides during the whole revolution of the crank. If the crank turns in a direction opposite to that indicated by the arrow in *Fig. 18½b* the crosshead will be pulled away from the slide, and this will often involve a more expensive construction of the crosshead or slide.

27.—For designing a connecting rod we generally use, as we have said before, empirical rules, such as will give proportions which will agree well with good practice and give the rod sufficient strength to resist all forces which at any time may act upon it. In order to show the application of these rules or formulas we take the following example.

EXAMPLE 6.—**It is required to compute the dimensions of a connecting rod and straps of the form**

shown in *Fig. 15b.* It is to be used for an engine having a cylinder 4 inches diameter; initial steam pressure, 121 pounds per square inch; stroke 6 inches; and the length of connecting rod is to be $2\frac{1}{2}$ times the stroke.

In accordance with the conditions given in the Example, the length of the connecting rod will be

$$2\tfrac{1}{2} \times 6 = 15 \text{ inches.}$$

Fig. 15b.—The diameter d at the neck is found by

RULE 1.—Divide the diameter in inches of the cylinder by 55 and multiply the quotient by the square root of the initial steam pressure per square inch on the piston; the result will be the diameter d. In symbols we have

$$d = \frac{D}{55} \times \sqrt{p}$$

in which d = diameter at the neck, D = diameter in inches of the cylinder and p the initial steam pressure in pounds per square inch on the piston.

Now, applying this rule, or what amounts to the same thing, substituting for the letters in the formula corresponding values given in the example and then performing the operation indicated by the signs, we have

$$d = \frac{4}{55} \times \sqrt{121} = \frac{4}{55} \times 11 = .8, \text{ say } \tfrac{25}{32} \text{ inch.}$$

For finding the diameter d_2 at the center of the rod we use

RULE 2.—Divide the diameter in inches of the cylinder by 50 and multiply the quotient by the square root of the initial steam pressure on the piston; the result will be the diameter d_2 in inches at the center of the rod. Or in symbols

$$d_2 = \frac{D}{50} \times \sqrt{p}$$

In which the letters, excepting d_2, denote the same quantities as given in connection with the preceding rule.

28.—The dimensions of the rectangular ends of the rod will depend on the dimensions of the crank pin and the crosshead pin. When these dimensions are known we may employ the following rules. Letters refer to *Fig. 15b.*

Rule 3.—Multiply the length of the crank pin by .7; the result will be the thickness of the rectangular end near the crank pin. In symbols we have

$$l \times .7 = t$$

in which l = length of crank pin, t = thickness of the rectangular end.

The reason for multiplying the length of the crank pin or crosshead pin by .7 is to give the brasses a good support. Some engineers use a slightly different multiplier to suit their individual judgment.

For the width w of the rectangular end we have,

Rule 4.—Add twice the thickness n of the brasses to the diameter of the crank pin; the sum will be the width w of the rectangular end near the crank pin. In symbols,

$$c + 2n = w$$

in which c = diameter of the crank pin, n = thickness of the brasses at the side of the pin, and w = width of the rectangular end.

For finding the dimensions of the rectangular end near the crosshead pin we use the same rules, remembering, of course, that in this case the letters l and c in the formula denote the length and diameter of the crosshead pin.

Let us assume that the length of the crank pin is $2\frac{1}{8}$ inches, and its diameter is $1\frac{3}{4}$ inches; the length of the crosshead pin is $1\frac{1}{4}$ inches and its diameter is 1 inch.

Applying Rule 3 we have for the thickness of the rectangular end near the crank pin,

$$l \times .7 = t$$
$$2.125 \times .7 = 1.48, \text{ say } 1\tfrac{1}{2} \text{ inches.}$$

The thickness n of the brass at the side of the crank pin, or that at the side of the crosshead pin in small engines, is usually $\frac{1}{4}$ inch; hence by Rule 4 we have for the width w of the rectangular end near the crank pin,

$$c + 2 \times n = w$$
$$1\tfrac{3}{4} + 2 \times \tfrac{1}{4} = 2\tfrac{1}{4} \text{ inches.}$$

For the thickness of the rectangular end near the crosshead pin we have by Rule 3,

$$l \times .7 = t$$
$$1.25 \times .7 = .875 = \tfrac{7}{8} \text{ inch.}$$

For the width w of the rectangular end near the crosshead we have by Rule 4,

$$c + 2 \times n = w$$
$$1 + 2 \times \tfrac{1}{4} = 1\tfrac{1}{2} \text{ inches.}$$

For finding the greatest thickness k of the brasses the following rule is generally employed:

Rule 5.—To $\frac{1}{8}$ of the diameter of the pin add $\frac{1}{4}$ inch; the sum will be the greatest thickness k of the brasses. In symbols,

$$\tfrac{1}{8}\, c + \tfrac{1}{4} \text{ inch} = k$$

in which c = diameter of the pin and k = greatest thickness of the brass.

In this example we have assumed that the diameter of the crank pin is $1\frac{3}{4}$ inches; hence by Rule 5 we find that the thickness k of the crank pin brasses will be

$$\frac{1}{8} \times c + \frac{1}{4} \text{ inch} = k$$
$$\frac{1}{8} \times 1.75 + \frac{1}{4} \text{ inch} = .46875 = \frac{15}{32} \text{ inch.}$$

The greatest thickness k of the crosshead pin brasses will be

$$\frac{1}{8} \times c + \frac{1}{4} = k$$
$$\frac{1}{8} \times 1 + \frac{1}{4} = \frac{3}{8} \text{ inch.}$$

To find the length of the rod from butt end to butt end we must deduct the distance m and m_2 from the length of the connecting rod. The distance m from the center of the crank pin to the butt end is equal to $\frac{1}{2}$ the diameter of the crank pin plus the greatest thickness k of the crank pin brass, which gives us

$$\frac{7}{8} + \frac{15}{32} = 1\frac{11}{32} \text{ inches.}$$

The distance m_2 from the center of the crosshead pin to the butt end of the rod will be

$$\frac{1}{2} + \frac{3}{8} = \frac{7}{8} \text{ inch.}$$

The sum of these two distances is equal to

$$1\frac{11}{32} + \frac{7}{8} = 2\frac{7}{32} \text{ inches.}$$

Now, subtracting this sum from the length of the connecting rod, which is 15 inches, we have

$$15 - 2\frac{7}{32} = 12\frac{25}{32} \text{ inches}$$

for the length l from butt end to butt end of the rod.

29.—For obtaining the dimensions of straps, cotters and gibs the student should employ the rules given in

Article 12 or use the formulas given in connection with *Fig. 9b.* In order to aid the students we will here compute the least thickness *b* of the strap at the crank pin end. All the reference letters refer now to *Fig. 9b.*

In order to determine this thickness we must first know the total pressure on the piston, which is

$$12.56 \times 121 = 1519.76 \text{ pounds.}$$

Allowing as before 5,000 pounds to the square inch for the safe working stress, we require an aggregate area in the cross-sections of both arms of the strap,

$$\frac{1519.76}{5000} = .3 \text{ inch,}$$

consequently, the area of the least cross-section at *b* through one arm of the strap must be

$$\frac{.3}{2} = .15 \text{ square inch;}$$

dividing this area by the width of the strap, which is equal to the thickness of the rectangular end, and which we have found in the preceding article to be $1\frac{1}{2}$ inches, we have

$$\frac{.15}{1.5} = .1 \text{ inch;}$$

adding $\frac{1}{8}$ inch we get

$.1 + .125 = .225$, say $\frac{1}{4}$ inch for the least thickness *b* of the strap.

It should be noticed that these computations are precisely the same as those given in Article 12, and the same remarks apply to the following computations.

For finding the least thickness b of the strap at the crosshead end, we proceed as before, thus:

We have found that the least cross-sectional area at b for resisting the stress on the strap must be .15 square inch.

In preceding article we have found that the thickness of the rectangular end of the rod near the crosshead pin must be $\frac{7}{8}$ inch; but this thickness is also the width of the strap; hence, dividing .15 by $\frac{7}{8}$ or its equivalent decimal .875 we have

$$\frac{.15}{.875} = .17, \text{ say } \tfrac{11}{64} \text{ inch};$$

adding $\frac{1}{8}$ inch we get $\frac{19}{64}$ inch for the least thickness b of the strap at the crosshead end. Now, remembering that the thickness $\frac{19}{64}$ is the unit of measurement for determining all the other dimensions of the strap at the crosshead end; and that $\frac{1}{4}$ is the unit of measurement for determining all the other dimensions of the strap at the crank pin end, there should be no difficulty in applying the formulas given in connection with *Fig. 9b* and thereby computing all the dimensions of the straps.

For the purpose of making this matter as plain as possible, we will briefly work out these dimensions, commencing with the strap at the crank pin end.

Fig. 9b.—For the thickness c we have in accordance with the formula given in connection with *Fig. 9b*,

$$c = 1\tfrac{3}{8}\, b$$

Using decimals in place of the fractions we have

$$c = 1.375 \times .25 = .34375 = \tfrac{11}{32} \text{ inch}.$$

For the thickness d at the crown of the strap

$$d = 1\tfrac{1}{2}b = 1.5 \times .25 = .375 = \tfrac{3}{8} \text{ inch.}$$

For the thickness e from the gib to the end of the strap,

$$e \times 2b = 2 \times .25 = .5 = \tfrac{1}{2} \text{ inch,}$$

and for the distance f

$$f = 1\tfrac{1}{2}b = 1.5 \times .25 = .375 = \tfrac{3}{8} \text{ inch.}$$

For the strap at the crosshead end we have

$$\begin{aligned} c &= 1\tfrac{3}{8}b = 1.375 \times .296 = .407000, \text{ say } \tfrac{13}{32} \text{ inch,} \\ d &= 1\tfrac{1}{2}b = 1.5 \times .296 = .4440, \text{ " } \tfrac{7}{16} \text{ "} \\ e &= 2b = 2 \times .296 = .592, \text{ " } \tfrac{19}{32} \text{ "} \\ f &= 1\tfrac{1}{2}b = 1.5 \times .296 = .4440, \text{ " } \tfrac{7}{16} \text{ "} \end{aligned}$$

For the width a and the thickness g of the cotter and gib at the crosshead end we have

$$\begin{aligned} a &= 1\tfrac{1}{4}B = 1.25 \times .875 = 1.09375 = 1\tfrac{3}{32} \text{ inch,} \\ g &= \tfrac{1}{4}B = .25 \times .875 = .21875 = \tfrac{7}{32} \text{ "} \end{aligned}$$

The taper of the cotter in this example is $\frac{5}{8}$ inch to the foot.

These proportions of cotter and gib are suitable for straps whose width B is equal to, or a very little larger than the diameter d at the neck of the rod. But when the former is much wider than the diameter at the neck, as is shown at the crank pin end, *Fig. 15½b*, the above proportions are not generally adopted. Very often in cases of this kind, the thickness of the cotter and gib is made equal to $\frac{1}{4}$ of the width of the strap; but whatever proportions may be adopted, the cross-sectional area at a of the cotter and gib should at least be equal to, not less than, the area at b of the thinnest part of the strap. It should also be remembered that the thickness $c = 1\frac{3}{8}b$ is suitable only when the thickness of the cotter is equal to $\frac{1}{4}$ of the width of the strap; when these proportions of the cotter are changed the thickness c must also be changed; in all cases the cross-sectional area of the strap through its slotted part should not be less than the cross-sectional area through the thinnest part of the strap.

Many engineers design the straps for connecting rods in large engines for a pull of $\frac{5}{8}$ of that on the connecting rod on each arm of the strap, and then add $\frac{1}{32}$ to $\frac{1}{16}$ inch for trueing up instead of $\frac{1}{8}$ inch as we have done in the preceding example. It may also be stated here that for computing the dimensions of the strap in the foregoing example, we have taken the pull on the piston rod instead of that on the connecting rod, but this will change the results to an inappreciable extent, for we have seen in Article 24 that the difference of the stresses in the piston rod and the connecting rod is so small that it may be neglected.

The whole result of our computations is shown in *Fig. 15½b*, which shows a drawing of a connecting rod agreeing with the conditions given in Example 6.

30.—*Fig. 16b* shows a connecting rod slightly different in form to the preceding one; it is tapered from d near the crosshead to d_2 at the center, and from the center d_2 to the crank pin end it is of uniform diameter. It is also designed for a strap at each end. For computing the dimensions of this rod we use the same rules as were used for computing the dimensions of the rod shown in *Fig. 15b*.

31.—*Fig. 17b* shows a connecting rod with a forked end for receiving the crosshead pin, which is fixed in the eyes of the fork by shrinking the latter on the crosshead pin, or fastening it by some other means, and thereby preventing the pin from turning in the eyes; the small

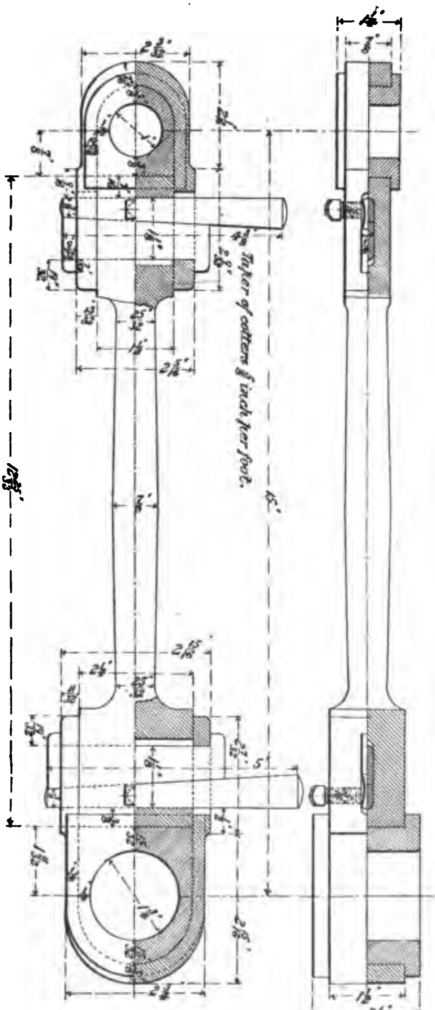


Fig. 15½b.

Connecting Rod.

Cylinder 4 inches diameter; stroke 6 inches.

Dimensions computed by rules in Articles 26b to 29b.

angular movement takes place in the brasses at the end of the piston rod or in the crosshead, as the case may be.

The rod has a uniform taper from d to c, but its thickness from d to e remains the same throughout; this is accomplished by cutting flats on its sides as shown.

The diameter d is found by Rule 1, Article 27, which, as we have seen, is expressed in symbols by the formula,

$$d = \frac{D}{55} \times \sqrt{p}$$

in which d = diameter of the rod at the neck; D = diameter in inches of the cylinder, and p = initial steam pressure per square inch of piston. The diameter c is equal to $1\frac{1}{2}$ times the diameter d.

These rules are so simple, and have been so fully explained in Article 27, that they require no further explanation.

It should be noticed that the crank pin end of this rod is designed for a strap. The hole a through the rectangular end is for the purpose of reducing the weight of the rod.

32.—*Fig. 18b* illustrates a connecting rod whose body is of a rectangular form. It is generally used in high speed engines. Its width has a uniform taper from b to f, but its thickness t is the same throughout. The area of the rectangular cross-section at t b of these rods should be equal to the area of the circular cross-section whose diameter is d in *Fig. 17b*; hence for computing the length of the sides b and t of the rectangular cross-section we assume that this rod has a circular cross-section at b and then compute the corresponding diameter d by the rule or formula,

$$d = \frac{D}{55} \times \sqrt{p}$$

as given in Article 27. When this diameter has been found we compute the corresponding area, and from this we calculate the length of the sides b and t of the rectangular section by the following rule:

Rule 6.—Divide the area by the number which indicates the number of times the width b has to be greater than the thickness t, and extract the square root of the quotient; the result will be the thickness of the rod. In symbols we have

$$t = \sqrt{\frac{a}{n}}$$

in which t = thickness of the rod, a = area of the cross-section at the neck. and n = the number by which the thickness of t must be multiplied for obtaining the width b.

Example 7.—**Compute the dimensions at b and t of the rectangular section of a connecting rod suitable for an engine with a cylinder 4 inches diameter, initial steam pressure 121 pounds per square inch of piston.**

Here we first assume that the cross-section at b is circular in form and then compute the corresponding diameter d by the formula

$$d = \frac{D}{55} \times \sqrt{p}$$

which gives us

$$d = \frac{4}{55} \times \sqrt{121} = \frac{4}{55} \times 11 = .8, \text{ say } \tfrac{13}{16} \text{ inch};$$

the area of a circle of this diameter is .518487 square inch.

The formula given with *Fig. 18b* shows that the width b is to be 1.5 times the thickness t; hence applying Rule 6, we get

$$\frac{.518487}{1.5} = .3456$$

Extracting the square root from this quotient we get

$$\sqrt{.3456} = .58, \text{ say } \tfrac{9}{16} \text{ inch}$$

for the thickness t.

The formula accompanying the illustration informs us that the width b should be 1.5 times the thickness t; hence the width b will be

$$b = 1.5 \times .5625 = .84375 = \tfrac{27}{32} \text{ inch.}$$

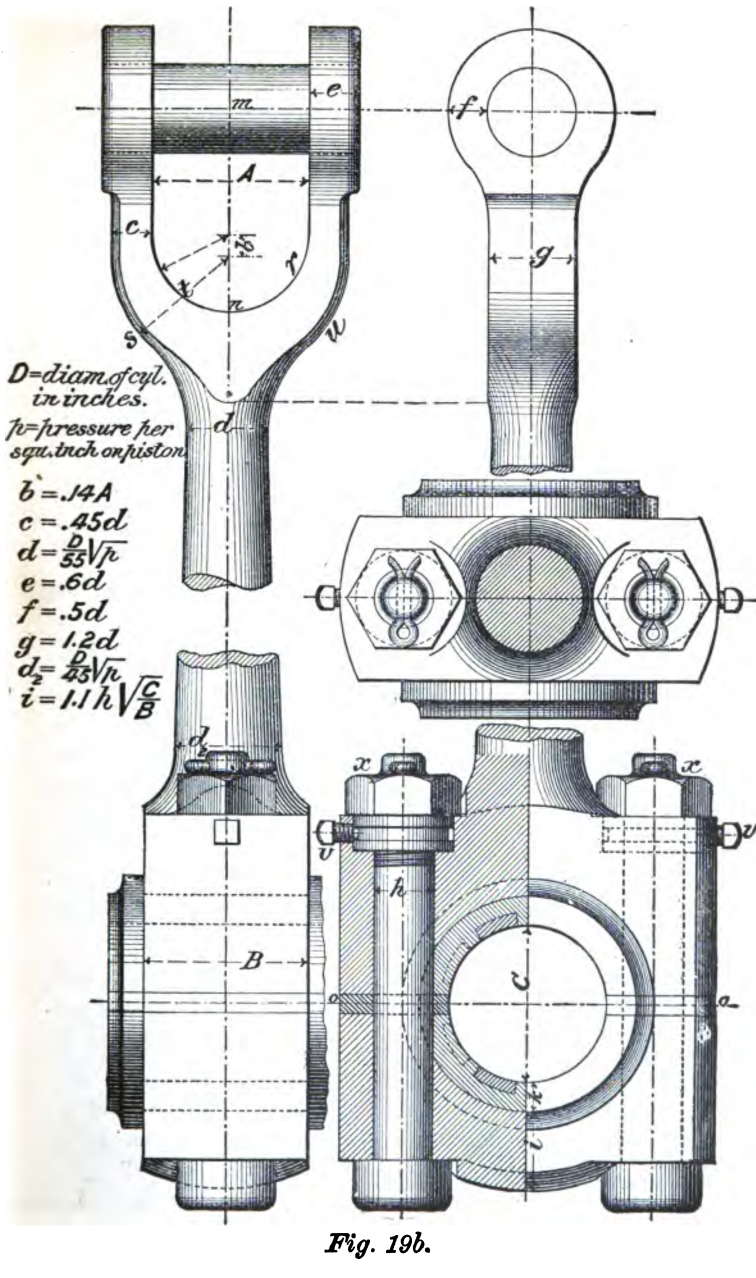
The width f near the crank pin end is to be 2.5 times the thickness t; hence for the width f we have

$$f = 2.25 \times .5625 = 1.265625 = 1\tfrac{17}{64} \text{ inches.}$$

The ends of this rod are designed for straps, and the dimensions of these are easily found by the rules given and explained in Article 29.

The hole a is for the purpose of reducing the weight of the rod.

33.—In regard to the forked end it should be said that the thickness $s\ t$, *Fig. 19b*, should be greater than at c, because the forces which the eyes have to resist will act with a leverage and increase the stresses near the section $s\ t$, and therefore the center from which the arcs s and u are drawn should be below the center from which the arc $r\ t$ is drawn, these centers should not coincide. The distance b between these centers should be .14 times A:



D=diam. of cyl. in inches.
p=pressure per squ. inch on piston
$b = .14A$
$c = .45d$
$d = \frac{D}{55}\sqrt{p}$
$e = .6d$
$f = .5d$
$g = 1.2d$
$d_2 = \frac{D}{45}\sqrt{p}$
$i = 1.1\,h\sqrt{\frac{C}{B}}$
m
e
f
A
c
b
r
t
n
s
u
g
d
d_2
x
v
h
B
C
o
i

Fig. 19b.

the distance A is that between the inner sides of the eyes as indicated on the drawing.

34.—It should also be noticed that in cases of this kind, where the pin is fastened in the eyes, the latter will be prevented from spreading and help to stiffen the forked end. But these conditions are changed in the forked end shown in *Fig 20b*; this forked end has no eyes; the arms of the fork support brasses in which the pin moves, and therefore it will have no tendency to prevent the arms of the fork from spreading; consequently, the section at $s_1 t_1$, *Fig. 20b*, must be made greater than at $s\ t$ in *Fig. 19b*; this is accomplished by increasing the distance b between the centers from which are drawn the arcs joining the branches of the fork to the rod, and in cases of this kind the distance b should be equal to $.24A$.

35.—The crank pin end of the rod in *Fig. 19b* has no strap, in place of which a cap is used, holding the brasses to the T end of the rod. With this construction the labor of finishing the rod and cap is simplified. The cap and rod are forged in one piece; the whole is turned and finished and the end bored out for the brasses, after which the cap is cut off, making the cut sufficiently wide for receiving the distance pieces $o\ o$; these are allowed to project beyond the inner surface of the iron, and thus prevent the brasses from turning in the rod.

36.—The form of the nuts $x\ x$ on the cap bolts is such as to prevent them from turning by means of the set-screws $v\ v$. The proportions of these nuts are given in *Fig. 362*, in "Meyer's Mechanical Drawing."

The dimensions of this rod are found by the rules or formulas given on the drawing, and are applied as shown in the following example:

EXAMPLE 8.—**Find the dimensions of a connecting rod of the form illustrated in *Fig. 19b*. It is to be suitable for an engine having cylinder 4 inches diameter, initial steam pressure 121 pounds per square inch of piston.**

The diameter d near the forked end is found by the rule or formula

$$d = \frac{D}{55} \times \sqrt{p}$$

in which d = diameter of the neck near the forked end; D = diameter in inches of the cylinder, and p = initial steam pressure per square inch of piston. The same formula was used before and explained in Article 27.

Substituting for the letters their respective values as given in the example, we have

$$d = \frac{4}{55} \times \sqrt{121} = \frac{13}{16} \text{ inch.}$$

For finding the diameter d_2 of the rod near the crank pin end we have a rule similar to the foregoing.

RULE 7.—Divide the diameter in inches of the cylinder by 45, and multiply the quotient by the square root of the initial steam pressure per square inch of the piston. In symbols

$$d_2 = \frac{D}{45} \times \sqrt{p}$$

in which the letters denote the same quantities as given in the preceding formula; this gives us

$$d_2 = \frac{4}{45} \times \sqrt{121} = .9777, \text{ say } \frac{31}{32} \text{ inch.}$$

The thickness c of the arm of the fork is to be .45 times the diameter d of the rod; hence we have

$$c = .45d = .45 \times .8125 = .365625, \text{ say } \tfrac{23}{64} \text{ inch.}$$

The thickness e of the eye is to be .6 times the diameter d of the rod; hence

$$e = .6d = .6 \times .8125 = .4875 = \tfrac{31}{64} \text{ inch.}$$

The width f of the eye is to be $\frac{1}{2}$ of the diameter d; hence

$$f = .5d = .5 \times .8125 = .40625 = \tfrac{13}{32} \text{ inch.}$$

The width g of the arms of the fork should be 1.2 times the diameter d; hence

$$g = 1.2d = 1.2 \times .8125 = .975 = \tfrac{31}{32} \text{ inch.}$$

37.—The cap at the crank pin end may be considered to be a beam uniformly loaded and supported at the centers of the bolts. If caps are used at the crosshead end, as shown in *Fig. 20b*, they will generally have oil holes; in such cases the diameter of the oil hole must be deducted from the width of the cap before the computation for the thickness at the center of the cap is made. Generally we can find the depth i of the cap, *Fig. 19b*, by the following rule:

To find the depth i of the cap at the crank pin end of a connecting rod.

Rule 8.—Divide the diameter C of the crank pin by the width B of the cap and extract the square root of the quotient; multiply this quotient by 1.1 and the diameter of the cap bolts; the result will be the depth i at the center of the cap. In symbols

$$i = 1.1 \times h \times \sqrt{\frac{C}{B}}$$

in which i = depth of the cap at its center C = diameter of the crank pin, and B = width of cap, h = diameter of the cap bolts.

For the caps at the crosshead end of a connecting rod, as shown in *Fig 20b*, we use the same rule, remembering that in this case the letter C in the above formula denotes the diameter of the crosshead pin and B the width of the cap, minus the diameter of the oil hole through the cap at the crosshead end.

EXAMPLE 9.—**Find the thickness i at the center of the cap on the crank pin end of a connecting rod of the form shown in *Fig. 19b;* the width of the cap is 1 1/2 inches, and the diameter of the crank pin 1 3/4 inches, diameter of bolts 5/8 inch.**

Substituting for the letters in the above formula the respective values given in the example we have

$$i = 1.1 \times .625 \times \sqrt{\frac{1.75}{1.5}} = .7423, \text{ say } \tfrac{3}{4} \text{ inch.}$$

38.—The bolts holding the cap to the end of the connecting rod should be made of mild steel and the nuts for these bolts should be made of wrought iron. Steel nuts on steel bolts are liable to seize and thereby subject one or the other to injury.

39.—It should also be remembered that bolts of comparatively small diameter are more liable to injury than larger ones in screwing up the nut. There may also exist an inequality of tension on the bolts due to screwing up. In order to allow for these undue stresses we must either choose a larger factor of safety or assume that the bolt is subjected to a greater pull than it actually is. The latter course is often adopted. We shall adopt the same course.

Since there are two bolts at the crank pin end of the rod which must resist the same amount of pull as there is on the connecting rod, each bolt must resist one-half the amount of the pull on the connecting rod; but for safety we shall assume that the pull on each bolt is two-thirds of that on the connecting rod, and find the diameter of the bolt which can resist this pull. Although the pull on the connecting rod is greater than that on the piston rod or of the total load on the piston, we should remember that this increase of pull on the connecting rod is so small that it can be neglected. Therefore in the following computations we shall assume that the pull on the connecting rod is equal to the total load on the piston. This will be made plainer by working out the following example.

EXAMPLE 10.—**Find the diameter of the bolts for securing the cap at the crank pin end of a connecting rod, suitable for an engine having a cylinder 4 inches in diameter, initial steam pressure 121 pounds per square inch of piston. The form of the connecting rod is to be like that shown in** *Fig. 19b*.

The area of a piston 4 inches in diameter is 12.56 square inches; consequently the total load on the piston will be

$$12.56 \times 121 = 1519.76 \text{ pounds;}$$

two-thirds of this is equal to

$$\tfrac{2}{3} \times 1519.76 = 1013.17 \text{ pounds,}$$

and this amount of pull each bolt must resist. The allowable working stress per square inch is 5,000 pounds; hence the area of the section at the bottom of the thread must be

$$\frac{1013.17}{5000} = .2026 \text{ square inch.}$$

Now, looking for this area at the bottom of the thread in a table of standard threads and bolts (an illustrated table of this kind will be found in Meyer's book on Mechanical Drawing), we find that two bolts ⅝ inch diameter will be required for holding the cap to the crank pin end of the connecting rod.

40.—Those portions of the body of the bolt which are not required for steadying the brasses, or distance pieces and cap for comparatively large connecting rods, are reduced in diameter and made equal to that at the bottom of the threads, as shown at the connecting rod end in *Fig. 20b*.

41.—*Fig. 20b* shows another connecting rod of the marine type, which is a more expensive one than the rod shown in *Fig. 19b* on account of the increase of the weight of metal in the brasses, particularly for large engines; these are also more expensive to renew when worn. The brasses are made in duplicate halves and are secured to the T end of the rod with two bolts and a cap, as shown.

The crosshead end of this rod is forked with a set of brasses similar in design to those at the crank pin, secured to each arm. In these brasses the journals of the crosshead pin have their small angular movement, and on account of this small angular movement the brasses should not be babbitted, for it has been observed that in such cases the babbitt metal is liable to cut and wear the journals oval. On the other hand, when a pin or shaft makes complete revolutions in its bearing the brasses should be babbitted as shown at the crank pin end; but there are some exceptions to this rule. For instance, when the journal is exposed to a great amount of dust, like those of the driving axle of a locomotive, it is best not to babbitt the bearings.

42 —At the crosshead end of this rod we have four bolts to resist the load on the piston, and therefore, theoretically, the bolts will be large enough when each can resist one-fourth of this load, but practically, particularly when small bolts are to be used, each bolt should be capable of resisting one-third of the load on the piston. For instance, let it be required to find the diameter of the bolts passing through the crosshead end of a connecting rod for an engine with a 4-inch cylinder, the initial steam pressure being 121 pounds per square inch of piston. In Article 39 we have found that the total load on this piston is 1519.76 pounds; one-third of this load is 506.59 pounds, and this is the pull which each bolt at the crosshead end must resist. Allowing 5,000 pounds per square inch for a safe working stress, we require

$$\frac{506.59}{5000} = .101 \text{ square inch}$$

for the area of the section at the bottom of the thread. Now, referring to a table of standard screw threads and bolts we find that the diameter of these bolts should be $\frac{1}{2}$ inch. On general principles we do not like to use bolts so small for any important work, and would therefore prefer to make these bolts $\frac{9}{16}$ inch in diameter.

43.—For determining the other dimensions of this connecting rod we use the formulas given with the illustration; these are practically the same as given in connection with *Fig. 19b*, and have been so thoroughly discussed in Article 36 that it will be unnecessary to repeat

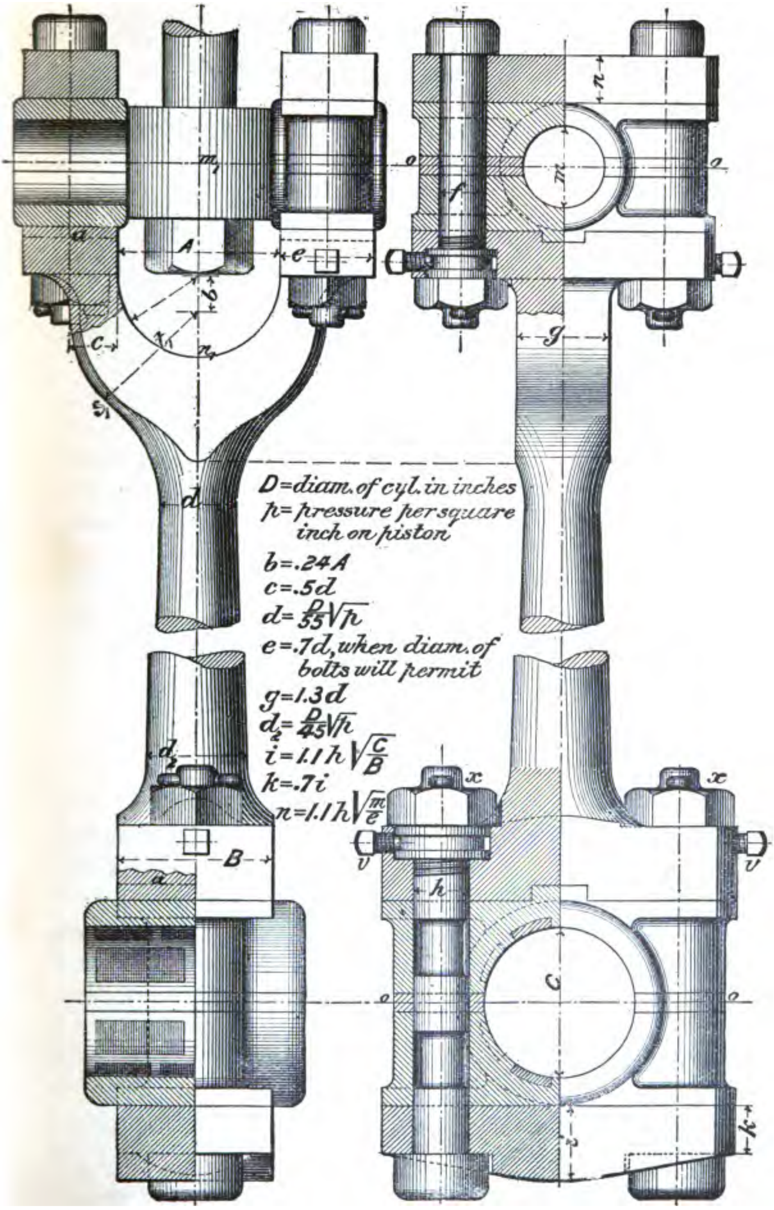

D=diam. of cyl. in inches
p= pressure per square inch on piston
$b = .24A$
$c = .5d$
$d = \frac{D}{55}\sqrt{p}$
e=.7d, when diam. of bolts will permit
$g = 1.3d$
$d_2 = \frac{D}{45}\sqrt{p}$
$i = 1.1h\sqrt{\frac{C}{B}}$
$k = .7i$
$n = 1.1h\sqrt{\frac{m}{e}}$

Fig. 20b.

here the whole operation for finding the dimensions of this rod.

Piston Rods.

44.—Piston rods may be made of steel or hammered wrought iron, and sometimes they are made of cold rolled iron; when made of the latter material they need no turning, and we have known these to give very good results.

In the direct acting type of engine, such as is shown in *Figs. 29b* and *30b*, the connecting rod is placed between the crank shaft and the cylinder, and therefore one piston rod only is needed for the piston. The diameter of a wrought iron piston rod may be found by the following rule.

Rule 9.—Divide the diameter in inches of the cylinder by 60 and multiply the quotient by the square root of the initial steam pressure in pounds per square inch of the piston; the result will be the diameter of the piston rod. In symbols we have

$$d = \frac{D}{60} \times \sqrt{p}$$

in which d = diameter of the piston rod, D the diameter in inches of the cylinder, and p the initial steam pressure in pounds per square inch of piston.

The diameter of a steel piston rod is found by the same rule, with this exception: Divide the diameter D of the cylinder by 69 instead of 60, and this will give us the following symbolic expression:

$$d = \frac{D}{69} \times \sqrt{p}$$

in which the letters denote the same quantities as in the preceding formula.

Piston rods are fastened to the piston in various ways. If the end of the rod is screwed into the piston, as shown in *Fig. 22b* then the diameter at the bottom of the thread of a wrought iron rod is found by

RULE 10.—Divide the diameter in inches of the cylinder by 80 and multiply the quotient by the square root of the initial steam pressure in pounds per square inch of the piston; the result will be the diameter at the bottom of the thread. In symbols

$$d_2 = \frac{D}{80} \times \sqrt{p}$$

in which the letters denote the same quantities as before.

For the diameter at the bottom of the thread of a steel piston rod use the same rule with this exception: Divide the diameter of the cylinder by 89 instead of 80. In symbols we have

$$d_2 = \frac{D}{89} \times \sqrt{p}$$

EXAMPLE 11.—**Find the diameter of a wrought iron piston rod, and also its diameter at the bottom of the thread; the diameter of the cylinder is 4 inches and the initial steam pressure 121 pounds per square inch of piston.**

According to Rule 9, we have

$$\frac{4}{60} \times \sqrt{121} = .066 \times 11 = .726, \text{ say } \tfrac{23}{32} \text{ inch.}$$

At the bottom of the thread the diameter will be

$$\frac{4}{80} \times \sqrt{121} = .05 \times 11 = .55, \text{ say } \tfrac{9}{16} \text{ inch.}$$

If the rod is to be made of steel, then its diameter would have to be

$$\frac{4}{69} \times \sqrt{121} = .058 \times 11 = .638, \text{ say } \tfrac{21}{32} \text{ inch,}$$

and at the bottom of the thread it will have to be

$$\frac{4}{89} \times \sqrt{121} = .045 \times 11 = .495 = \text{say } \tfrac{1}{2} \text{ inch.}$$

If the piston rod is to be of cold rolled iron, then its diameter may be equal to that of a steel one.

45.—In the return connecting rod type of engines, in which the crank shaft is placed between the crosshead and the cylinder, two piston rods for a piston are necessary, as shown in *Fig. 23b.* If the rods are of wrought iron, we find the diameter d by Rule 10 which, as we have seen, is expressed in symbols by

$$d = \frac{D}{80} \times \sqrt{p}$$

For computing the diameter at the bottom of the thread of this rod the same rule is used, with this exception: Divide the diameter of the cylinder by 100 instead of 80; or in symbols

$$d_2 = \frac{D}{100} \times \sqrt{p}$$

EXAMPLE 12.—**Find the diameter of a wrought iron piston rod, two rods being used for the piston; the diameter of the cylinder is 4 inches, and the initial steam pressure 121 per pounds per square inch.**

$$d = \frac{D}{80} \times \sqrt{p} = \frac{4}{80} \times \sqrt{121} = .05 \times 11 = .55 = \text{say } \tfrac{9}{16} \text{ inch.}$$

At the bottom of thread the diameter will be

$$d_2 = \frac{D}{100} \times \sqrt{p} = \frac{4}{100} \times \sqrt{121} = .04 \times 11 = .44. \text{ say } \tfrac{7}{16} \text{ inch.}$$

For small engines the piston rod is generally screwed into the piston, as shown in *Fig. 22b*, because when the diameter d of the rod and the diameter of the threaded part has been established by the foregoing rules it will be found that the difference of these diameters is so small as to prevent a taper sufficiently large for drawing the rod out of the piston without much difficulty. Of course the end of the rod could be made of equal diameter throughout; but since the rod must have a good tight fit into the piston it becomes very laborious work to take the rod out when this becomes necessary. It is for this reason that a tapered end is adopted, and for an easy withdrawal of the rod a taper, that is, a variation in the diameter of $\frac{3}{4}$ inch per foot, at least, should be chosen; and if such a taper cannot be used throughout the whole depth of the hub of the piston it should extend as far as the minimum diameter of the

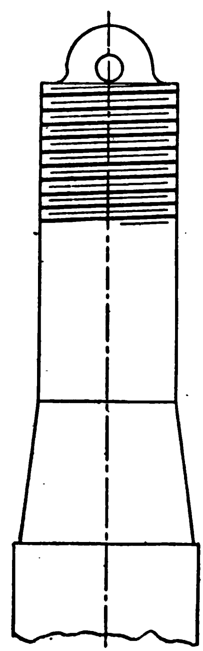


Fig. 21½b.

end will allow, and the remainder should be made cylindrical, or parallel, as it is technically called. In cases of this kind the taper is often much greater than given above, so that the end of the piston rod will have the appearance shown in *Fig. 21½b.*

Even with a larger taper difficulty will, in time, be encountered in removing the rod unless there is a shoulder of $\frac{1}{16}$ inch left on the rod for small engines, or $\frac{1}{8}$ for larger engines. The shoulder will also prevent a wedging action tending to split the piston; it also allows the rod to be returned without incroaching on the taper, when this becomes necessary after a long wear.

When a piston is cottered to the piston rod, as shown in *Fig. 25b*, then the area of the weakest section at *b* should not be less than ½ the area of the section at *d* of the rod.

The dimensions of this cotter can be found by the formulas given with *Fig. 6b.*

46.—The diameters of the piston rods as found above are suitable only when the length of the rods does not exceed 20 times its diameter. For rods whose length is 30 times the diameter use the following formula.

$$d = \frac{D}{48} \times \sqrt{p}$$

instead of the one given in connection with Rule 9.

47.—For a medium stroke oscillating engine we may use the formula

$$d = \frac{D}{45} \times \sqrt{p}$$

The letters in both of these formulas denote the same quantities as given in Article 44.

There is not much danger of the nuts working loose at the end of the piston rod; but it will be better for the sake of safety to put a pin through the end as shown. It may also be stated that the use of the nut at the end of the threaded part of the piston rod is optional; in some engines it is not used, as will be seen by referring to *Fig. 30b.*

Pistons.

48.—One of the important requirements of a piston is that it shall be steam tight during its motion from end to end of the cylinder, and work with as little friction as possible. Another requirement is that it must be strong enough, with the minimum amount of metal, to resist the pressure which may act upon it, due allowance being made for the shocks to which a piston is sometimes subjected. On these requirements, and on some minor considerations, the form of the piston depends.

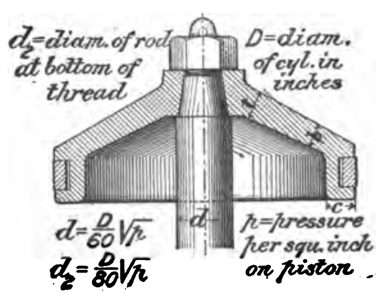


Fig. 21b.

Fig. 22b shows a dished piston with single plate. With this type not only is the weight reduced, but it also has the advantage of allowing the stuffing box of the piston to extend further into the cylinder, and enables us to reduce the length of the piston rod.

The dimensions of this piston may be found in the following manner. First find the thickness t of the plate near the hub by

Rule 11.—Divide the diameter of the cylinder in inches by 150 and multiply the quotient by the square root of the initial steam pressure in pounds per square inch of piston; the result will be the thickness t near the hub of the piston. In symbols

$$t = \frac{D}{150} \times \sqrt{p}$$

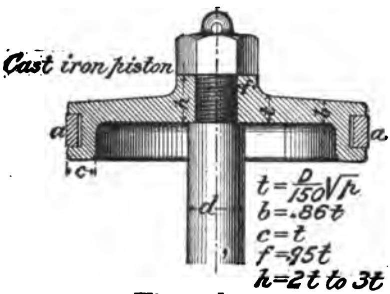


Fig. 22b.

in which t = thickness of metal near the hub, D = diameter in inches of the cylinder, p = initial steam pressure in pounds per square inch.

EXAMPLE 13.—**Find the thickness t of the plate of the piston near its hub for a cylinder 4 inches diameter, initial steam pressure 121 pounds per square inch.**

Substituting for the letters in the formula their respective value given in the example we have

$$\frac{4}{150} \times \sqrt{121} = .0266 \times 11 = .2926 = \text{say } \tfrac{19}{64} \text{ inch.}$$

The thickness b of the plate near its circumference is 86 times the thickness t, or

$$b = .86 \times t = .86 \times .2968 = .255, \text{ say } \tfrac{1}{4} \text{ inch.}$$

The width c of the flange at the bottom of the packing ring is equal to the thickness t; hence it is $\frac{19}{64}$ inch.

The thickness f of the metal in the hub at the end of the rod may be .95 times the thickness t; hence

$$f = .95 \times t = .95 \times .2968 = .28196, \text{ say } \tfrac{9}{32} \text{ inch.}$$

The depth h of the hub may be from 2 to 3 times the thickness of t.

In designing a piston care should be taken not to have too great a difference in the thickness of its metal, because, if this difference is too great a variation in the contraction of the metal may take place and set up stresses in the piston which may ruin its usefulness.

49.—*Fig. 24b* shows an outside view of the piston with the joint of the packing ring in plain view.

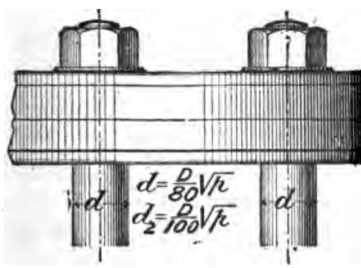


Fig. 23b.

Fig. 26b illustrates the method of making the joint of the packing ring. A solid ring is turned $\frac{1}{8}$ inch larger in diameter than that of the cylinder; notches *a* and *b* are then cut on each side of the ring, each extending exactly through one-half the width of the ring, and slightly over three times as long as the original diameter of the ring is greater than the diameter of the cylinder; the ends of the notches must meet each other so as to cut the ring apart. The ends are then pressed together as shown in *Fig. 24b*, and a pin driven through them so as to hold them in this position while the ring is being turned to the diameter of the cylinder; the pin is then removed and the piston with packing ring placed in the cylinder; the inherent elasticity of the ring will press it against the walls of the cylinder and keep the piston steam tight.

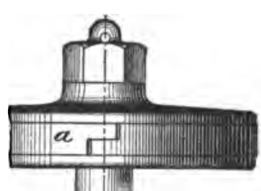


Fig. 24b.

The width of the packing ring may be equal to about one-half of the diameter of the piston rod, and the thickness of the ring may be equal to about one-half its width. However, these proportions of the packing ring depend much on the individual preference of the designer; some engineers make the width of the ring greater than we have given and make its cross-section square. When the thickness of the ring is changed the dimension of *c* must also be changed.

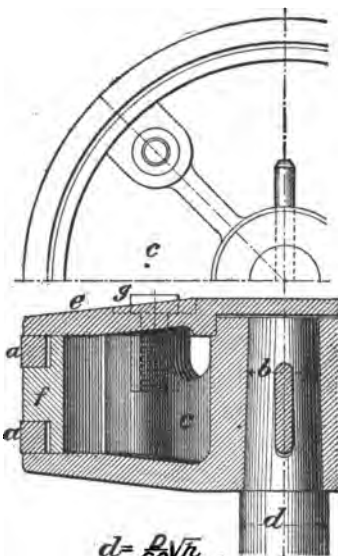


Fig. 25b.

Sometimes the single plate pistons are coned, as shown in *Fig. 21b.* In this type the slope of the outer surface should be about 1 in 3. When this piston is made of cast iron, the thickness of the metal may be determined by the same formulas given with *Fig. 22b*; if it is to be made of steel then use the formulas given with *Fig. 28b.*

50.—In each of the pistons previously illustrated only one cast iron packing ring is used, and no provisions are made for an easy and safe way of placing the ring into position or for taking it out. The only way for placing a ring of this kind in the grooves of the piston is to force the ends apart where it has been cut so that it can be passed over the body of the piston, but in doing so there is a liability of breaking the ring, particularly when its diameter is small.

Fig. *25b* shows a piston with two cast iron packing rings and with provisions for an easy and safe way of putting them into position and removing them. This piston consists of a body *c*, frequently called a spider, a cast iron follower plate *e* secured to the body by the follower bolts *g*, the cast iron T ring *f*, and the cast iron packing rings *a a*.

The T ring is solid (not cut). Before the follower plate is put on, it is slipped over the arms of the body which hold it in position, and when the follower plate is put on, two grooves on the circumference of the piston will be formed for the reception of the packing rings, the lower one being put on before the T ring is slipped over the body; after the T ring has been placed in position the upper ring is put in and the whole secured by the follower plate. Of course in fitting up the piston care must be taken not to clamp fast the packing rings; they must have a steam tight fit and be free to move to adjust themselves to the walls of the cylinder. In a similar easy and safe way the packing rings can be removed.

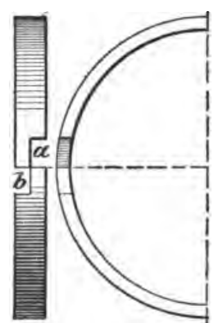


Fig. 26b.

If the follower bolts are screwed directly into the cast iron of which the body is made there will be a liability of rust causing considerable trouble in removing them; to obviate this, gun metal or brass plugs are first screwed tightly into the body, and these are tapped for receiving the follower bolts.

51.—*Fig. 27b* shows a cast iron hollow piston strengthened by internal radial ribs. The dimensions of this piston have been taken from one in successful service and gives us a good opportunity of comparing a few of these dimensions with those obtained from the following rules.

52.—The diameter of the follower bolts may be found by the following rule.

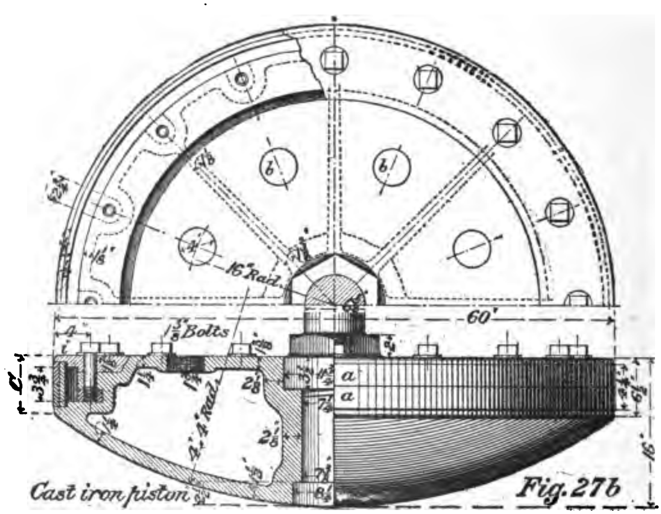


Fig. 27b.

RULE 12.—Divide the diameter of the cylinder by 400 and multiply the quotient by the square root of the initial steam pressure in pounds per square inch of piston and add $\frac{1}{4}$ inch to the product; the result will be the diameter of the follower bolts. In symbols,

$$d = \frac{D}{400} \times \sqrt{p} + \tfrac{1}{4} \text{ inch},$$

in which d = diameter of the follower bolt, D = the diameter in inches of the cylinder and p = the initial steam pressure in pounds per square inch.

EXAMPLE 14.—**Find the dimensions of follower bolts for a piston 60 inches diameter, initial steam pressure 49 pounds per square inch.**

Substituting for the letters in the formula their respective values given in the example, we have

$$d = \frac{60}{400} \times \sqrt{49} + \tfrac{1}{4} \text{ inch} = .15 \times 7 + \tfrac{1}{4} \text{ inch} = \text{say } 1\tfrac{5}{16} \text{ inch}.$$

EXAMPLE 15.—**Find the diameter of the follower bolts for a piston 60 inches diameter initial steam pressure 121 pounds per square inch.**

$$d = \frac{60}{400} \times \sqrt{121} + \tfrac{1}{4} \text{ inch} = .15 \times 11 + \tfrac{1}{4} \text{ inch} = \text{say } 1\tfrac{7}{8} \text{ inch}.$$

EXAMPLE 16.—**Find the diameter of the follower bolts for a piston 18 inches diameter, initial steam pressure 121 pounds per square inch.**

$$d = \frac{18}{400} \times \sqrt{121} + \tfrac{1}{4} \text{ inch} = .045 \times 11 + \tfrac{1}{4} \text{ inch} = \tfrac{3}{4} \text{ inch}.$$

The pitch, that is the distance from center to center of the follower bolts, may be from 7 to 10 times their diameter.

In *Fig. 27b* the follower bolts are not screwed into the metal of the body of the piston; they have gun metal nuts placed in the recesses in the piston.

Three or four holes in the follower through which the bolts pass should be tapped so that eye bolts can be screwed into them for the purpose of lifting the follower off the body of the piston.

The number of ribs in this piston may be found by the following rule:

Rule 13.— Divide the diameter in inches of the piston by 10 and add 2; the result will be the number of ribs in the piston. In symbols

$$n = \frac{D}{10} + 2$$

in which n = number of ribs and D = diameter in inches of the piston.

Example 17.—**What will be the number of ribs in a piston 60 inches diameter?**

$$n = \frac{60}{10} + 2 = 8 \text{ ribs.}$$

The thickness of the ribs may be found by the following rule:

Rule 14.—Divide the diameter in inches of the piston by 300 and multiply the quotient by the square root of the initial steam pressure in pounds per square inch; the result will be the thickness of the ribs. In symbols

$$t = \frac{D}{300} \times \sqrt{p}$$

in which t = thickness of the ribs, D = diameter in inches of the piston, and p = the initial steam pressure per square inch.

EXAMPLE 18.—**What will be the thickness of the ribs in a piston 60 inches diameter, initial steam pressure 49 pounds per square inch?**

$$t = \frac{60}{300} \times \sqrt{49} = .2 \times 7 = 1{,}4 = \text{say } 1\tfrac{13}{32} \text{ inch.}$$

The holes b b in the upper side of the piston provide means for taking the core out of the piston, and when this has been done they are tapped and cast iron plugs screwed in.

53.—The depth of the piston at the center can be found by the following rule:

RULE 15.—Divide the diameter in inches of the piston by 36 and multiply the quotient by the square root of the initial steam pressure in pounds per square inch; add $1\frac{1}{2}$ inches to the product; the result will be the depth of the piston at the center. In symbols

$$\frac{D}{36} \times \sqrt{p} + 1\tfrac{1}{2} \text{ inches,}$$

in which D = diameter of the piston and p = initial steam pressure in pounds per square inch.

EXAMPLE 19.—**Find the depth at the center of a piston 60 inches diameter, initial steam pressure 49 pounds per square inch.**

Substituting for the letters in the formula their respective values given in the example, we have

$$\frac{60}{36} \times \sqrt{49} + 1\tfrac{1}{2} \text{ inches} = 1.666 \times 7 + 1\tfrac{1}{2} \text{ inches} = \text{say } 13\tfrac{5}{32} \text{ inches.}$$

The breadth C of the ring face, *Fig. 27b*, may be .1 to .15 times the diameter of the cylinder. In symbols

$$C = .1 \times D \text{ to } .15 \times D$$

in which C = breadth of ring face, and D = diameter of the cylinder.

Cast iron is well adapted for hollow pistons such as shown in *Fig. 27b*. In these the requisite stiffness can always be secured by giving the proper depth to the piston and stiffening its faces by radial ribs, but such a construction makes a piston heavier than is desirable; for this reason a form of piston shown in *Fig. 28b* is at present very often used.

54.—*Fig. 28b* shows a steel piston made of a single thickness of metal and dished to a conical form so as to secure the necessary rigidity without any ribs. A piston

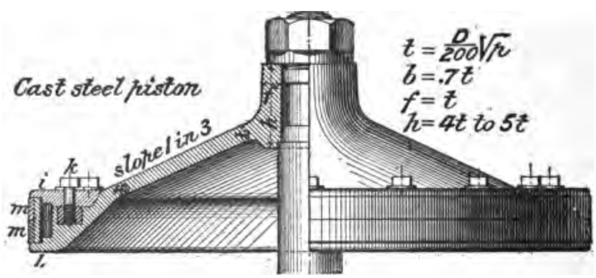


Fig. 28b.

of this kind is much lighter than the one shown in *Fig. 27b*. The proportions of this piston are indicated by the formulas accompanying the drawing. These formulas are handled in precisely the same manner as explained in the preceding Articles, and it will be unnecessary to take up space here for explaining them.

When these pistons are to be made of exceptionally good metal which has been melted at least twice, the thickness of metal can be a little less than obtained by the formulas; but if a poorer quality of metal is used, these thicknesses should be a little greater than obtained by the computations.

The construction for the packing rings is the same in both pistons, shown in *Fig. 27b* and *Fig. 28b*. Two cast iron packing rings are used in each piston with a single cast iron ring placed inside of the packing rings. All these rings are cut, so that they can adjust themselves to the cylinder walls.

In order to avoid leakage the cut of the inner ring should not be placed directly behind the cut of either packing ring. The springs for pressing the rings against the cylinder walls are placed in the space behind the inner ring. These springs may be half elliptical, or of the Z form. Some builders use spiral springs placed in pockets cast in the body of the piston.

The inner ring has a tendency to equalize the pressure of the springs and cause the packing rings to act with a comparatively uniform pressure against the walls of the cylinder.

For upright engines the springs are placed all around the body of the pistons. For horizontal engines, the springs at the bottom of the piston are replaced by solid blocks which support the weight of the piston.

55.—*Figs. 29b* and *32b* show elevations and *Fig. 30b* shows a vertical section of a small stationary engine with cylinder 4 inches diameter and 6 inches stroke, designed for a steam pressure of 121 pounds per square inch. In this design all the working parts are easily accessible

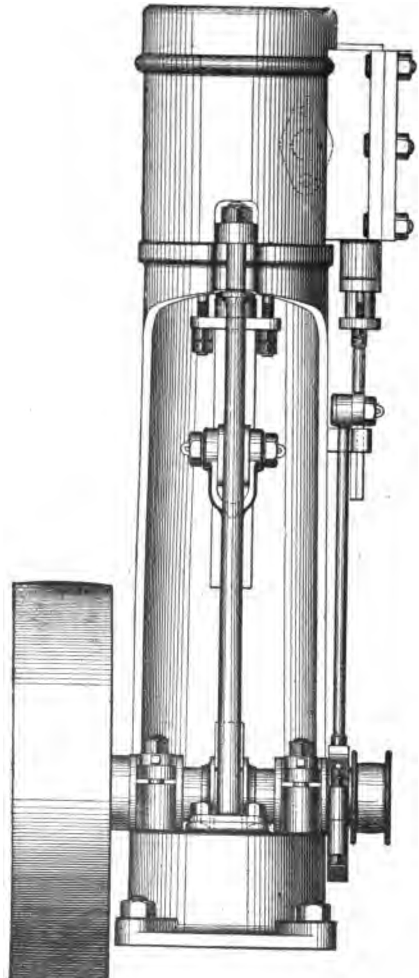


Fig. 29b.

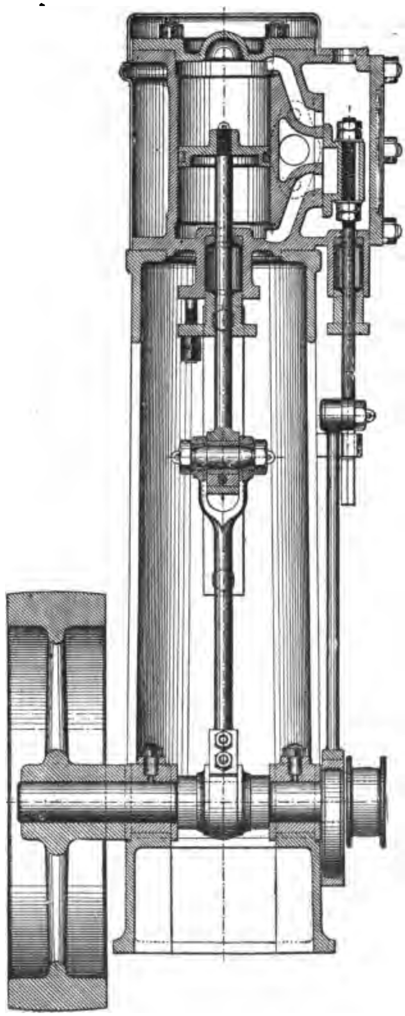


Fig. 30b.

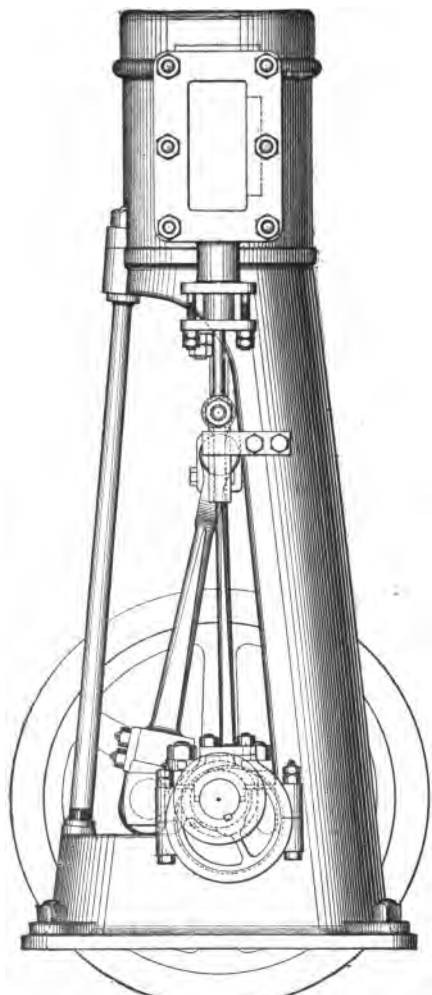


Fig. 32b.

for oiling; they are in plain sight, so that anything wrong in the action of the different parts can be readily detected.

In making a drawing of this engine the eccentric must be shown in its correct position, and since this is generally an exceedingly interesting subject to the mechanic, involving very important principles, useful to know in setting the eccentric on the crank shaft, we will here consider the construction of the simple valve gear.

Slide Valve.

56.—The function of the slide valve is to govern the proper distribution of the steam in the cylinder, and on this the form of the slide valve and the setting of the eccentric depends. We shall first consider the form of the slide valve.

57.—*Fig. 34b* shows a section of the slide valve, of the steam passage $S\ S_1$, and the exhaust passage E. The surface ab on which the valve slides is called the "valve face." The openings in the valve seat in which the steam passages terminate are called the "steam ports," and the opening in the valve seat in which the exhaust passage terminates is called the "exhaust port." A steam chest surrounds the slide valve; but in these detail diagrams the steam chest is only partially shown. The valve seat should always be raised a little above the surrounding metal and the ends of the valve should travel a little beyond the valve seat.

The partitions $d\ d$ (*Fig. 34b*) which separate the steam passages and the exhaust passage are called "bridges." The steam passages have a double duty to perform: First, they must conduct the steam from the steam chest to the cylinder in which the steam performs its work, and

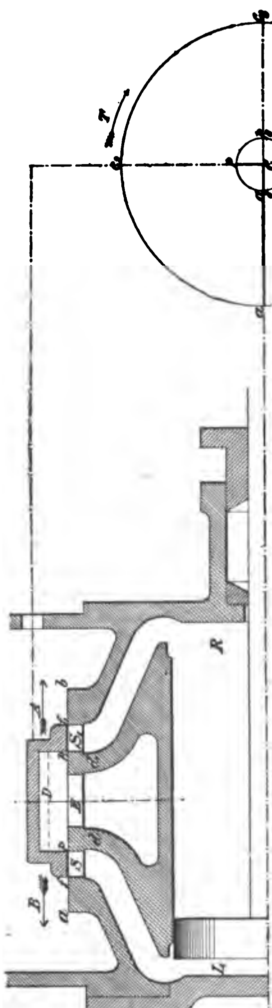


Fig. 33b.

after the work has been done the steam passages must conduct the steam from the cylinder to the exhaust cavity in the slide valve, from whence it flows through the exhaust passage into the air, providing it is a non-condensing engine.

58.—For the admission of steam we do not require so great a steam port as is needed for the release of steam. Hence, if the steam ports are large enough for the exhaust there will be ample provisions for the admission of steam. In high speed engines, for instance, locomotives, there is seldom any trouble in getting the steam into the cylinder, but sometimes a difficulty arises in getting the steam out of the cylinder rapidly enough. Consequently, in some engines the valve does not open the steam ports to their full width for the admission of steam, and thus reduce the travel of the valve. The friction between the valve and its seat is considerable, and by reducing the travel of the valve, power is saved.

59.—*Fig. 33b* shows the slide valve midway of its travel, or in its central position, and in this position it covers the steam ports, no more and no less. A valve of this kind is said to have no steam lap nor inside lap. It should also be noticed that when this valve is in its central position no steam can enter or leave the cylinder; but if the valve is moved in the slightest degree in either direction, say in the direction of the arrow *A*, then steam will be admitted to the left hand end *L* of the cylinder, and at the same time the exhaust of the steam from the right hand end *R* of the cylinder will commence. It may also be said that with this valve steam will be admitted to the cylinder during the whole length of the stroke.

60.—*Fig. 34b* shows a section of a slide valve with its flanges $f f_1$ overlapping the steam ports when it stands in its central position. The amount l which each flange of the valve overlaps the steam port is called "outside lap," "steam lap," or simply "lap." Here we would caution the student not to fall into an error and apply the term lap to the total amount of overlap at both ends of the valve; the lap is always measured at one end of the valve when it stands in its central position. If the amount l which the valve overlaps the steam port S measures $\frac{7}{8}$ inch, then the valve has $\frac{7}{8}$ inch lap. The effect of lap is that the admission of steam will cease before the piston reaches the end of its stroke, or we may say that the steam is cut off before the end of the stroke.

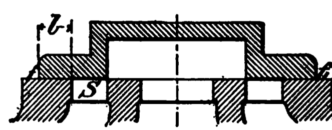


Fig. 34b.

61.—*Fig.* 35*b* shows a section of a slide valve whose exhaust edges $p\ p_1$ overlap the inner edges of the steam ports. The amount i which each of these edges overlap the inner edge of the steam port is called "inside lap" or "exhaust lap." The exhaust lap is always less than the steam lap, and in many cases it is not used.

Sometimes the valve is constructed so that the inner edges $p\ p_1$ will control the admission of steam, and the outer edges $f\ f_1$ will control the exhaust; in such cases the inner lap is the steam lap and the outer lap is the exhaust lap. From this it will be seen that the terms steam lap and exhaust lap are preferable to the terms outside lap and inner lap. The former terms will in some cases avoid ambiguity.

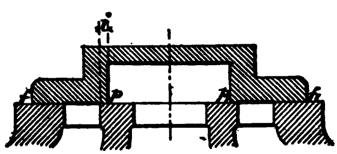


Fig. 35*b*.

62.—*Fig.* 36*b* shows a section of a slide valve whose inner edges $p\ p_1$ do not cover the steam ports. The opening r, when the valve is in its central position, is called "inside clearance" or "negative exhaust lap."

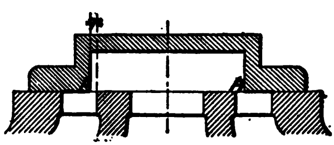


Fig. 36*b*.

Fig. 40*b*.—"Lead" is the amount of opening k of the steam port when the piston is at the beginning of its stroke. If the amount of opening k is $\frac{1}{16}$ inch when the piston is at beginning of its stroke, the valve is said to have $\frac{1}{16}$ inch lead. In modern horizontal engines the valve has practically no lead. In upright engines the lead is generally a little greater at the bottom of the cylinder than at the top.

63.—All slide valves must fulfill three conditions:

First—Steam must not be admitted to both ends of the cylinder at the same time;

Second—The exhaust should commence not later than the admission of steam at the other end of the cylinder;

Third—The outer edges of the steam ports must be covered completely when the valve is in its central position, so that steam cannot pass from the steam chest into the exhaust port, as shown in *Fig. 37b.*

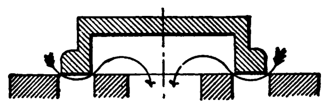


Fig. 37.

It will be seen that all the slide valves shown in *Fig. 34b*, *35b* and *36b* will fulfill these conditions; but the valve shown in *Fig. 37* will not, and therefore should not be used.

64.—The slide valve receives its motion from the eccentric which is keyed to the crank shaft. The eccentric and rod convert circular motion into a reciprocating motion, that is, to-and fro straight-line motion, and their action is exactly the same as that of the crank and connecting rod, with this exception, the latter converts the reciprocating motion into a circular motion.

Eccentric.

65.—There is nothing mysterious about an eccentric, as some mechanics seem to think. The eccentric may be considered to be a small crank with a very large crank pin, thus:

Fig. 38b.—Let *C* be the center of the crank shaft and *e* the center of the eccentric disk. We may now consider the distance *Ce* to be the length of the crank arm, and the diameter *a* of the disk to be the crank pin diameter, which is so large as to take in the diameter of the shaft. Now, since the diameter of the crank pin can have no effect on

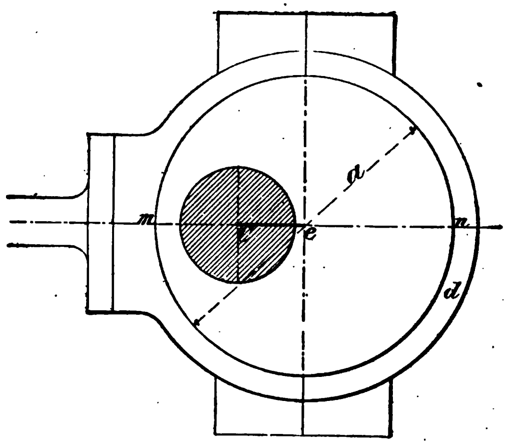


Fig. 38b.

the resulting reciprocating motion, it follows that this motion will be precisely the same as that derived from a crank. The reason for using an eccentric is that there will be no necessity for cutting or dividing the crank shaft.

66.—The distance from the center *C* of the shaft to the center *e* of the eccentric is called the "throw of the eccentric." When the slide valve and the eccentric are connected by a rod, as shown in *Fig. 32b*, then the throw is equal to one-half tne travel of the valve.

Some authorities define "throw" as equal to twice the distance Ce; in such a case the throw will be equal to the travel of the slide valve. We mention this so as to avoid confusion; nearly all the modern writers employ the first definition, and we shall use the same in this book.

67.—The line Ce produced in both directions to m and n is called the "center line of the eccentric."

68.—For engines in which the slide valve and eccentric are connected, as shown in *Fig. 32b*, the throw of the eccentric is equal to the lap of the slide valve plus the amount of opening of the steam port for the admission of steam. If, for instance, the slide valve has a $\frac{1}{2}$ inch lap, and the width of the steam port is $\frac{9}{16}$ inch, which is to be opened fully and no more, then the throw of the eccentric will be equal to $\frac{1}{2} + \frac{9}{16} = 1\frac{1}{16}$ inches, and the travel of the valve will be $2\frac{1}{8}$ inches. If the slide valve is to open the steam port to $\frac{3}{8}$ inch only for the admission of steam, then the throw will be equal to $\frac{1}{2} + \frac{3}{8} = \frac{7}{8}$ inch, and the travel of the valve will be $1\frac{3}{4}$ inches. Other kinds of connections may affect the amount of throw considerably, as is explained in "Meyer's Mechanical Drawing and Machine Design."

69.—In studying the effect produced on the distribution of steam by the lap given to the slide valve we may obtain a better conception of the whole subject by the use of a model of the valve gear, so that the relative positions of the piston and the slide valve can readily be seen. But models of this kind are expensive, and in the absence of a model we will have to work out these positions on the drawing board; but as soon as we attempt to do so we are confronted by a complication due to the obliquity of the connecting rod and the eccentric rod. The obliquity of the latter is so small that generally it may be neglected;

but that of the connecting rod has quite an effect on the position of the piston, and to this we shall now direct attention.

70.—*Fig. 39b.*—Let *C* be the center of the crank shaft and let the horizontal line *Ab*, which passes through the center *C*, be the center line of motion of the crosshead, piston rod and piston. Through the point *C* draw a line *Cp* perpendicular to the line *Ab* and from the point *C* as a center and with a radius equal to the length of the crank draw a circle *apbm*; this circle is called the "crank pin circle," and represents the path of the centre of the crank pin. It cuts the perpendicular *Cp* in the point *p*, and cuts the horizontal line *Ab* in the points *a* and *b*. These two latter points are called the "dead centers," because when the crank pin is on either one of these points the connecting rod will have no tendency to turn the crank shaft in one or the other direction; and if the engine should stop with the crank pin on one of these dead centers it cannot be started without employing some external force beside that exerted by the connecting rod.

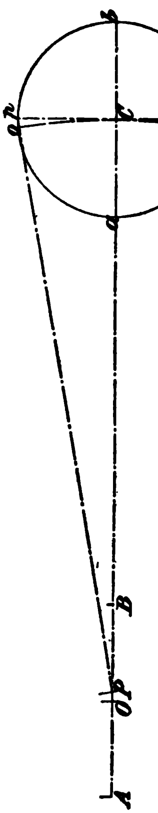


Fig. 39b.

From the point *a* as a center and with a radius equal to the length of the connecting rod draw a short arc cutting the line Ab in the point A; this point will be the center of the crosshead pin at one end of its travel. From the point b as a center and with a radius equal to the length of the connecting rod draw a short arc cutting the line Ab in the point B; this point will be the center of the crosshead pin at the other end of its travel. Since the crosshead and the piston are rigidly connected by the piston rod we may take the distance from A to B to be the stroke of the piston, and this is equal to the diameter ab, which is equal to twice the length of the crank.

From p as a center and with a radius equal to the length of the connecting rod draw a short arc cutting the line Ab in the point P; this point will be the position of the center of the crosshead pin when the crank pin is at p. Now notice: The point p bisects the semicircumference which extends from a to b; but, on the other hand, the point P does not bisect the stroke AB; it will be found that the point P is closer to the end B of the stroke than it is to A.

Let O be the center of the stroke; from this point as a center and with a radius equal to the length of the connecting rod draw the arc oCm terminating in the circumference of the crank pin circle; the two points o and m will be the centers of the crank pin when the piston is at the center of its stroke. From this we see that during the first half stroke of the piston the crank pin travels from a to o, and during the second half stroke of the piston the crank pin travels from o to b. Since the motion of the crank pin should be uniform, the average speed of the piston during the the first half stroke will be a little greater than during the second half stroke; and this difference in the speed of the piston is due to the obliquity of the connecting rod. In the same way it may be shown that the obliquity of the eccentric rod has an effect on the motion of the slide valve; this, however, is very small, and may be neglected. In an attempt to trace the relative motions of the slide valve and piston, these complications may lead to confusion and considerable trouble. We state here, briefly, the result produced by the obliquity of the connecting rod is an unequal cut-off, or, in other words, the steam will not be cut off at equal distances from the ends of the cylinder,

71.—This inequality of the cut-off can be reduced by increasing the length of the connecting rod, and if we assume this length to be infinite, the inequality of the cut-off will disappear altogether; but for the study of the valve motion a better way will be to throw out of consideration the connecting and eccentric rods, and assume the slide valve stem and the piston rod to be connected to the crank shaft by slotted crossheads.

Position of the Eccentric for a Valve without Lap.

72.—In order to give the valve the correct motion for the desired distribution of steam it is of the highest importance to place the eccentric in the correct position on the crank shaft. If the slide valve has no lap and no lead the center line of the eccentric must be perpendicular to that of the crank, as indicated by the heavy center lines Cp and aC in the diagram *Fig. 33b.* Again, in all cases the eccentric must travel ahead of the crank; hence if the direct connections between the valve and the eccentric are used, that is, if there is no rocker placed between them, then the center line of the eccentric must be above the center line of the crank, as shown in *Fig. 33b.* The correctness of this statement is seen by an inspection of this figure. When the piston commences to move towards the right hand end *R* of the cylinder the valve must move in the direction of arrow *A* so as to open the steam port *S* and admit steam behind the piston for driving it in the desired direction.

If we place the center line of the eccentric on the dotted line *Cg* and then expect the crank shaft to turn in the direction of arrow *T* we shall be doomed to disappointment, because an eccentric placed in this position will cause the valve to move in the direction of the arrow *B* when the crank shaft is turned in the direction of arrow *T*, consequently the valve will not open the steam port *S* for the admission of the steam required for driving the piston to the other end of the cylinder. But if we turn the crank in a direction opposite to that of the arrow *T*, then the valve will open the steam port *S* and admit steam behind the piston and drive it in the desired direction. From this it is plainly seen that the eccentric must be placed in advance of the crank in the direction of its motion. If the eccentric is placed above the crank as shown in *Fig. 33b*, then the shaft will rotate in the direction of the arrow *T*; if placed below the crank it will turn in the opposite direction.

73.—Now let us trace the movement of the slide valve during a complete revolution of the crank; the valve has no lap or lead. We have seen that in *Fig. 33b* the piston is at the beginning of its stroke and the slide valve occupies its central position, as it should do.

After the crank has advanced through an angle of 90 degrees, the center line of the crank will be at Cc_2, and the center line of the eccentric will occupy the position Cp_2. During this advance of the crank the piston has travelled from the beginning to the center of its stroke, and the valve has moved from its central position to the end of its travel, thereby opening the steam port S for the admission of steam to the space behind the piston, and opening the steam port S_1 for the release of the steam in front of the piston.

After the crank has advanced through an additional angle of 90 degrees the center line of the crank will be at Cc_3, and the center line of the eccentric will be at Cg. During this advance of the crank the piston has reached the end of its stroke and the valve has been travelling in the direction of arrow B, gradually shutting off the supply of steam behind the piston, and also closing the release or exhaust of steam in front of the piston, and thus reaches its original central position, as shown in the illustration.

During the time that the crank travels from Cc_3 to Cg the eccentric advances from Cg to Ca_2, and the valve moves in the direction of the arrow B from its central position to the end of its travel, and thus opens fully the steam port S_1 for the admission of steam to the right hand end R of the cylinder and drives the piston on its return stroke; it also opens the steam port S for the release of steam in front of the piston. During the last quarter of the revolution of the crank the valve moves in the direction of arrow A from the end of its travel to its original central position.

In tracing the motion of this valve we must not lose sight of the fact that we are dealing with a valve which has no lead or lap, and we will notice that the steam port S is open during the whole stroke of the piston from L to R for the admission of steam, and the steam port S_1 is open during the same whole stroke of the piston for the exhaust of steam.

Position of the Eccentric for a Valve with Lap.

74.—Now let us see what effect the lap and lead of the valve will have. In the first place, the lap and lead will change the position of the eccentric on the crank shaft. In *Fig. 40b* the piston is at the beginning of its stroke, and steam will be required for driving the piston, hence the

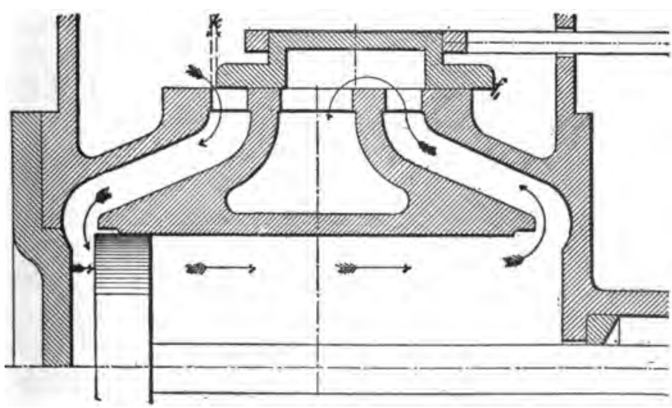


Fig. 40b.

valve must open the steam port S as soon as motion commences, or, in many cases, the valve has lead, as shown. Consequently, the valve cannot stand in its central position at the beginning of the stroke; it must be moved ahead to an amount equal to the lap and lead, and this cannot be done without shifting the eccentric. In the preceding Article we have seen that when the valve has no lap or lead the center line of the eccentric must stand perpendicular to the center line of the crank, or we may say the advance of the eccentric must be equal to an angle of 90 degrees.

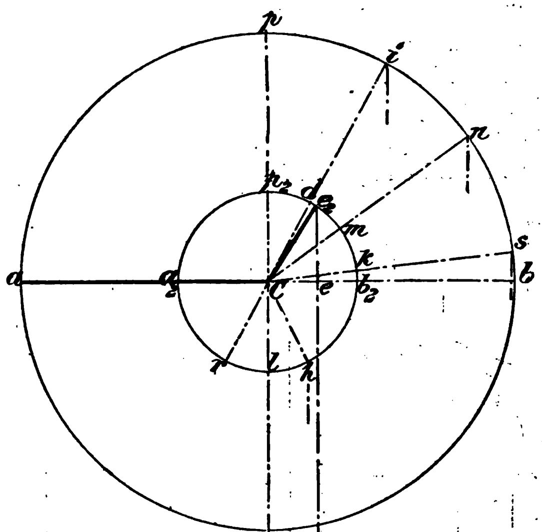


Fig. 41b.

But now we see that when the slide valve has lap and lead the advance of the eccentric must be greater than an angle of 90 degrees. This additional advance is found as follows:

75.—*Fig. 41b.*—Draw the horizontal center line *ab*, and from any point *C* on this line as a center draw the eccentric circle $a_2 p_2 b_2$ cutting the vertical line *Cp* in the point p_2. From the center C lay off on the line ab a distance Ce equal to the lap and lead; through the point e draw a line ee_2 perpendicular to ab cutting the eccentric circle in the point e_2; this point will be the center of the eccentric; join the the points e_2 and C by a straight line. The angle p_2Ce_2 is called the "angle of advance," and the distance Ce is called the "linear advance." Probably a practical example will make the whole subject clearer.

Example.—Find the position of an eccentric for a valve which has $\frac{1}{2}$ inch lap and $\frac{1}{16}$ inch lead. The steam ports are $\frac{5}{8}$ inch wide and should be opened fully for the admission of steam; stroke 6 inches. The connection between the eccentric and the slide valve is the same as that shown in *Fig. 32b.*

The first step will be to find the throw of the eccentric. In Article 68 we have seen that for a direct connection between the valve and the eccentric, such as we have to deal with now, the throw of the eccentric is equal to the sum of the lap and the steam port opening; hence the throw will be $\frac{1}{2} + \frac{5}{8} = 1\frac{1}{8}$ inches.

The radius of the eccentric circle is equal to the throw, hence from any point C, *Fig. 41b*, as a center and with a radius of $1\frac{1}{8}$ inches draw the eccentric circle $a_2b_2p_2$; also from the same center draw the crank pin circle abp; draw the horizontal line ab cutting the crank pin circle in the points a and b; the line ab will represent the stroke of the piston, and aC the center line of the crank at the beginning of the stroke. From the center C lay off on the line ab a distance equal to $\frac{9}{16}$ inch (which is the sum of the lap and lead) and thus establish the point e; through this point draw the line ee_2 perpendicular to the line ab

cutting the eccentric circle in the point e_2; this point will be the center of the eccentric.

76.—At the beginning of the stroke the center line of the connecting rod coincides with the center line of motion of the piston, and therefore the position of the eccentric as found above is correct, no matter what obliquity the connecting rod may have during the stroke. We have now worked out all the conditions given in the example.

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